PB85-180875

Performance Evaluation Guide for Large Flow Ventilation Systems

PEDCo-Environmental, Inc., Cincinnati, OH

Prepared for

Environmental Protection Agency
Research Triangle Park, NC

May 84

U.S. Department of Commerce
National Technical Information Service
NTIS

United States
Environmental Protection
Agency

Office of Air Quality
Planning and Standards
Research Triangle Park NC 27711

EPA-340/1-84-012
July 1984

Air

PB85-180875

EPA

Performance Evaluation Guide For Large Flow Ventilation Systems

REPRODUCED BY
NATIONAL TECHNICAL
INFORMATION SERVICE
U.S. DEPARTMENT OF COMMERCE
SPRINGFIELD, VA. 22161

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before completing)

1. REPORT NO.

2.

3. RECIPIENT'S ACCESSION NO. PB85 180875 /AS

4. TITLE AND SUBTITLE
Performance Evaluation Guide for Large Flow
Ventilation Systems

5. REPORT DATE May 1984

6. PERFORMING ORGANIZATION CODE

7. AUTHOR(S)
W. F. Kemner, R. W. Gerstle, and Y. M. Shah

8. PERFORMING ORGANIZATION REPORT NO. 3760-1-119

9. PERFORMING ORGANIZATION NAME AND ADDRESS
PEDCo Environmental, Inc.
11499 Chester Rd.
Cincinnati, OH 45246

10. PROGRAM ELEMENT NO.

11. CONTRACT/GRANT NO.
68-01-6310
Work Assignment 119

12. SPONSORING AGENCY NAME AND ADDRESS
U. S. Environmental Protection Agency
Control & Process Engineering
Stationary Source Compliance Division
Washington, D.C.

13. TYPE OF REPORT AND PERIOD COVERED Final

14. SPONSORING AGENCY CODE

15. SUPPLEMENTARY NOTES
Project Officer for SSCD: Dwight Hlustick

16. ABSTRACT

Air pollution control systems in the primary metals industry, particularly the steel and copper segments, rely on large capture and ventilation systems with flow rates commonly in the range of 50,000 to 1,000,000 acfm and greater. These systems are used primarily to control process fugitive emissions from various furnaces and for building evacuation.

Because these systems are in integral feature of the compliance programs of the industries involved, this manual was initiated to accomplish the following:

* To provide design and operation and maintenance guidelines to state and local agency personnel who evaluate the performance of these systems.
* To provide a comprehensive treatment of the existing literature with regard to technical and specific aspects of typical designs.
* To provide an easy-to-read technical manual on design and operation for the use of inspectors.

Inasmuch as ventilation systems are highly complex from a design standpoint and experience plays a major role in most designs, this manual should be considered an introductory primer rather than a detailed design manual.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS	b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Ventilation	Fans, Ducts, Hoods	13A
Industrial Engineering	Steel Industry	13H
	Copper Smelting	

18. DISTRIBUTION STATEMENT Unlimited	**19. SECURITY CLASS** *(This Report)* Unclassified	**21. NO. OF PAGES** 146 p.
	20. SECURITY CLASS *(This page)* Unclassified	**22. PRICE** A07

EPA Form 2220-1 (Rev. 4-77) PREVIOUS EDITION IS OBSOLETE

EPA-340/1-84-012

Performance Evaluation Guide For Large Flow Ventilation Systems

by

William Kemner
Richard Gerstle
Yatendra Shah

Contract No. 68-01-6310
Work Assignment No 119

EPA Project Officer: John Busik
EPA Project Manager: Dwight Hlustick

U.S. ENVIRONMENTAL PROTECTION AGENCY
Office of Air Quality Planning and Standards
Stationary Source Compliance Division
Washington, DC 20460

May 1984

1a

DISCLAIMER

The information in this document has been funded wholly or in part by the United States Environmental Protection Agency under Contract No. 68-01-6310 to PEDCo Environmental, Inc. It has been subject to the Agency's peer and administrative review, but it does not necessarily reflect the views of the Agency and no official endorsement should be inferred.

CONTENTS

Page

Figures v
Tables viii
Acknowledgment ix

1. Introduction 1

2. General Ventilation and Hooding Principles 3

    Design basis 5

3. Hood Design Considerations 11

    Determining air flow requirements 16

4. Duct Design and Considerations 33

    Transport velocities 34
    Energy losses 36
    Branched systems 43

5. Fan Systems 46

    Fan types and operating characteristics 46
    Forced versus induced draft 55
    Fan requirements for emission control system applications 57
    Fan arrangements 60
    Fan drives 60
    Fan controls 61
    Fan sizing 66

6. Ventilation System Inspection 71

    Preparing for inspection 71
    Safety considerations 72
    Onsite company-inspector interaction 73
    Inspection procedures 73
    Operation and maintenance (O&M) considerations 79

CONTENTS (continued)

Page

7. Total Furnace Enclosures 85

Electric arc furnaces 85
Basic oxygen furnaces 95

8. Special Applications 103

Coke oven sheds 103
Electric arc furnace ventilation 107
Blast furnace casthouse control 115
Control systems on basic oxygen furnaces (BOF's) 117
Building evacuation 123
Copper converters 124

References 128

Appendix A - Bibliography A-1

FIGURES

Number		Page
1	Matrix of Hooding Applications for Particulate Control	4
2	Example of Open and Closed Hooding on the Discharge End of a Sinter Strand	6
3	Direct Shell Evacuation on an Electric Arc Furnace	8
4	Variation in Gas Flow Rate From a BOF During the Course of a Heat	10
5	Conveyor Transfer Point Hooding Using Total Enclosure	12
6	Exterior EAF Canopy Hood	14
7	Effect of Excessive Plume Velocity	17
8	Velocity Contours (Expressed in Percentage of Opening Velocity) and Streamlines for Circular Openings	19
9	Velocity Contours and Streamlines for Flanged Hood	19
10	Formulas for Estimating Hood Air Flows	20
11	Dimensions Used to Design High-Canopy Hoods for Hot Sources	21
12	Schematic Arrangement of Ladle Hood for Reladling Emission Control	25
13	Controlled Airflow From a Heated Source	28
14	Uncontrolled Airflow From a Heated Source	28
15	General Principle of the Push-Pull (Air-Curtain) Type System	29
16	Comparison of Face Velocity Decay for Blowing Versus Exhausting	31

FIGURES (continued)

Number		Page
17	Air Curtain Control System on a Capped Converter	32
18	Dew-Point of Air Containing Various SO_3 Concentrations	35
19	Pressure Measurements in Ducts	38
20	Simple Pressure Diagram	42
21	Taper Duct System	44
22	Centrifugal Fan Components and Layout of a Typical Industrial Fan System	47
23	Centrifugal Fan Blade Configurations and Impeller Arrangements	48
24	Fan Testing Procedure and Typical Characteristic Curves	50
25	Typical Characteristic Curves for a Backward-Curved-Blade Centrifugal Fan	51
26	Typical Characteristic Curves for a Forward-Curved-Blade Fan	53
27	Typical Characteristic Curves for a Straight-Blade Fan	54
28	Basic Principle of Induced Versus Forced Draft	56
29	Louvre Damper: (a) Parallel Blade Multilouvre; (b) Opposed Blade Multilouvre; (c) View of Parallel Blade, Multilouvre Damper View Showing Linkage	63
30	Guillotine Damper: (a) Simplified Cross-Sectional View of a Guillotine Damper; (b) Guillotine Isolation Damper Using Seal Air; (c) Top-Entry Type Guillotine Damper, Showing Operation	64
31	Butterfly Damper: (a) Simplified Cross-Sectional View of a Butterfly Damper; (b) Butterfly Damper Showing Hand Operator	65
32	Typical Furnace Enclosure	87
33	Furnace Enclosure at North Star Steel Company	91
34	Furnace Enclosure at Birdsboro Corporation	93

FIGURES (continued)

Number		Page
35	Typical BOF Furnace Enclosure	97
36	Schematic of Basic Oxygen Secondary Emission Control System of Kaiser Steel-Fontana	98
37	Schematic of Q-BOP Secondary (Charging) Emissions Control System of Republic Steel, Chicago	100
38	Various Shed Configurations	104
39	Thermal Expansion of Hot Gases From a Push	105
40	Electric Arc Furnace Utilizing Partial Enclosure	108
41	Ventilation Systems for Electric Arc Furnaces	110
42	Combined Direct Shell Evacuation With Canopy Hood	111
43	Multiple Pickup Points Vented to Common Control Device	112
44	Flow Rate Required for Electric Arc Furnace Control	114
45	BOF Hood Arrangements	119
46	Canopy Hood Concept for BOF Charging Emissions	121
47	Gaw Damper (Closure Plate) Use in BOF Control	122
48	Pierce-Smith Converter	125
49	Secondary Converter Hood Configuration	126

TABLES

Number		Page
1	Comparison of Principal Data for 10, 30, and 100 Percent Combustion in BOF Hooding	9
2	Range of Capture Velocities	15
3	Range of Design Velocities	37
4	Conversion Table for Duct Velocity to Velocity Pressure	39
5	Relative Advantages of Using Duct Dampers	45
6	Basic Fan Laws	58
7	Basic Damper Types	62
8	Air Density Correction Factor	68
9	Density of Common Gases	69
10	Data on EAF Plants Designed for Total Furnace Enclosure	94
11	Flow Balancing of a Typical Furnace Enclosure With Additional Pickup Hoods, Connected to a Common Baghouse	95
12	BOF/Q-BOP Shops Utilizing Furnace Enclosures	101
13	Example of Flow Balancing of Multiple Evacuation System on Electric Arc Furnace	113
14	Blast Furnace Casthouse Typical Volume Requirements	116

ACKNOWLEDGMENT

This report was prepared for the U.S. Environmental Protection Agency by PEDCo Environmental, Inc., Cincinnati, Ohio. Mr. Dwight Hlustick was the EPA Project Officer. Mr. William Kemner served as the PEDCo Project Manager. The principal authors were Mr. Kemner and Messrs. Richard Gerstle and Yatendra Shah. Messrs. Gopal Annamraju, Gary Saunders, and Lario Yerino prepared specific sections of the report. The authors wish to thank Mr. Hlustick for his overall guidance and direction on this task.

SECTION 1

INTRODUCTION

The purpose of this manual is to familiarize Agency inspectors with the design principles and O&M considerations for large-scale (i.e., generally >50,000 cfm) ventilation systems commonly found in the metallurgical industry. The emphasis is on the steel industry because most large, complex systems are found in the many individual processes used in this industry. Applications in copper smelting and other industries are also discussed. Inasmuch as ventilation systems are highly complex from a design standpoint and experience plays a major role in most designs, this manual should be considered an introductory primer rather than a detailed design manual.

Several standard publications discuss ventilation principles and fan engineering.[1-4] In general, however, these publications emphasize smaller, more traditional applications. Furthermore, their emphasis is on ventilation in the general sense, as opposed to air pollution control. The complicating factors found in ventilation of large metallurgical systems are either treated in the abstract or not at all. Literature on operation and maintenance practices and inspection procedures is very limited.

Air pollution control systems in the primary metals industry, particularly the steel and copper segments, rely on large capture and ventilation systems with flow rates commonly in the range of 50,000 to 1,000,000 acfm and greater. These systems are used primarily to control process fugitive emissions from various furnaces and for building evacuation.

Because these systems are an integral feature of the compliance programs of the industries involved, this manual was initiated to accomplish the following:

- To provide inspection and operation and maintenance guidance to state and local agency personnel who evaluate the performance of these systems.

- To provide a comprehensive treatment of the existing literature with regard to technical and specific aspects of typical designs.
- To provide an easy-to-read technical manual on design and operation for the use of inspectors.

Although not treated in detail in this manual, other technologies are available for control of process fugitive emissions in the metallurgical industries. These technologies, which do not involve hooding and ventilation systems, include roof-mounted electrostatic precipitators[5,6] and fume suppression utilizing inert gases to suppress the oxidation of molten metal.[7] Both of these technologies offer promise of lower cost than conventional ventilation approaches. They have been applied for control of charging and tapping emissions in steel making plants and control of blast furnace casting emissions. The former have not yet found application in the United States.

Sections 2 through 5 present technical factors of design for hooding, ducting, and fans. Section 6 describes inspection procedures for use in assessing the effectiveness and maintenance of ventilation systems. Section 7 deals with total furnace enclosures. In Section 8, the foregoing information is supplemented in the context of special problems that are found in several specific applications. The appendix contains a bibliography for those interested in pursuing the subject matter further.

SECTION 2

GENERAL VENTILATION AND HOODING PRINCIPLES

Process hooding and ventilation systems are required to capture and transport emissions to some control device or vent. These hooding and ventilation systems sometimes also eliminate potential industrial hygiene problems by reducing employees' exposure to an air contaminant and by removing heat from the process area. Figure 1 summarizes the various types of hooding used in processes in the metallurgical industries.

The three basic parts of a ventilation system are the hood or air intake, for initial capture of the emissions; the ductwork, for transport of the gas stream to the vent or control device; and a fan, to move the gas stream. Whereas the design of the basic hood and ventilation system is well understood for small and medium-sized systems, the application of the same principles to large processes often results in marginal or inadequate systems, especially when high-temperature processes are involved. This inability to apply the same principles results primarily from the large size of the equipment, the high heat loads, the variability of conditions in batch processes, the need for access to the process, and greater maintenance requirements. For larger systems, much of the design is left to the ingenuity and experience of the designer, who must fit the hood around the process and lay out the ductwork with minimal interferences.

Inadequate design of a ventilation system can compromise overall performance. In all cases, the hood must be sized and oriented to capture the maximum quantity of emissions without requiring excessive gas volumes (a trade-off between performance and energy consumption). It makes little sense to install a high-efficiency control device if a major portion of the emissions are not captured initially. The hood should be as close as possible to the point of generation without interfering with equipment movement and process operation. It should be oriented to minimize cross-drafts and to take advantage of thermal drafts.

	Application	Fixed hoods	Movable hoods	Air curtain/ push-pull systems	Enclosures	Curtain walls	Runner covers	Roof canopies	Ladle hoods	Close-fitting hoods	Side-draft hoods	Telescoping hoods	Sheds
		Type of hooding used											
Ferrous	Coke oven pushing	X	X							X		X	X
	Basic oxygen furnace blowing	X			X					X			
	Basic oxygen furnace fugitives	X			X				X				
	Electric arc furnace refining			X	X	X		X		X	X		
	Electric arc furnace fugitives			X	X	X		X					
	Open hearth tapping		X						X				
	Blast furnace casting	X	X	X		X	X					X	
	Blast furnace slag granulation	X				X							
	Steel scarfing	X								X			
	Grinding and shotblasting	X								X			
	Hot rolling operations	X											
	Sinter strand discharge	X			X					X			
	Hot metal transfer	X	X						X	X			
	Raw metal conveyors	X			X					X			
	Gray iron cupola		X				X						
Nonferrous	Primary copper converters	X	X	X									
	Primary copper reverberatory furnaces						X		X				
	Primary lead blast furnace		X				X						
	Primary aluminum electrolytic furnace	X	X		X								
	Casting shakeout system	X			X								
Other	Minerals handling	X			X							X	
	Minerals crushing	X			X								

Figure 1. Matrix of hooding applications for particulate control.

The ductwork leading from the hood (or pickup point) to the control device must be sized to provide the needed transport velocity--generally between 15 and 25 m/s (2700 and 4500 ft/min) --depending on particulate loading and size distribution. Layout of ductwork should minimize energy losses caused by bends, transitions, branches, etc., and should also minimize air inleakage. If the source is hot, refractory lining or water-cooled hoods and ducts may be required.

The three basic types of process hooding and venting systems are close-fitting hoods, canopy hoods, and so-called building evacuation. (The latter term is used loosely because in very few cases is an entire building actually evacuated.) More than one of these three systems may be utilized on a single process.

2.1 DESIGN BASIS

The most important of the three ventilation system components (i.e., the hood, the duct, and the fan) is the hood. The ventilation system will not perform well unless the hood effectively captures the emissions. The hood design and open face area determine the amount of air that is drawn into the system to capture the emissions. The volume of air and the process emissions then determine the size of the ductwork, and these factors and the pressure drop required by the control device in turn determine the size of the fan. To minimize capital cost and fan power requirements, the designer tries to minimize the amount of outside air drawn into the system. A face velocity of 200 to 500 ft/min is usually required through the hood's open area. Thus, to minimize total ventilation air requirements, the hood must fit closely to the process and have a small open area. The inability to achieve these goals is the major problem in the applications discussed in this manual. Figure 2 illustrates the enclosure principle applied to a hood on the discharge end of a sinter strand. In this application, nearly total enclosure is possible.

Under some conditions, the hood may be fitted directly to the process [(e.g., direct shell evacuation (DSE) on an electric arc furnace (EAF) in which the furnace roof serves as the hood)]; this arrangement allows only the process gases (and minimal air infiltration) to pass through the vent system. Even in these cases, however, high temperatures and explosive gases must be considered. The gases must be transported at concentrations that are less

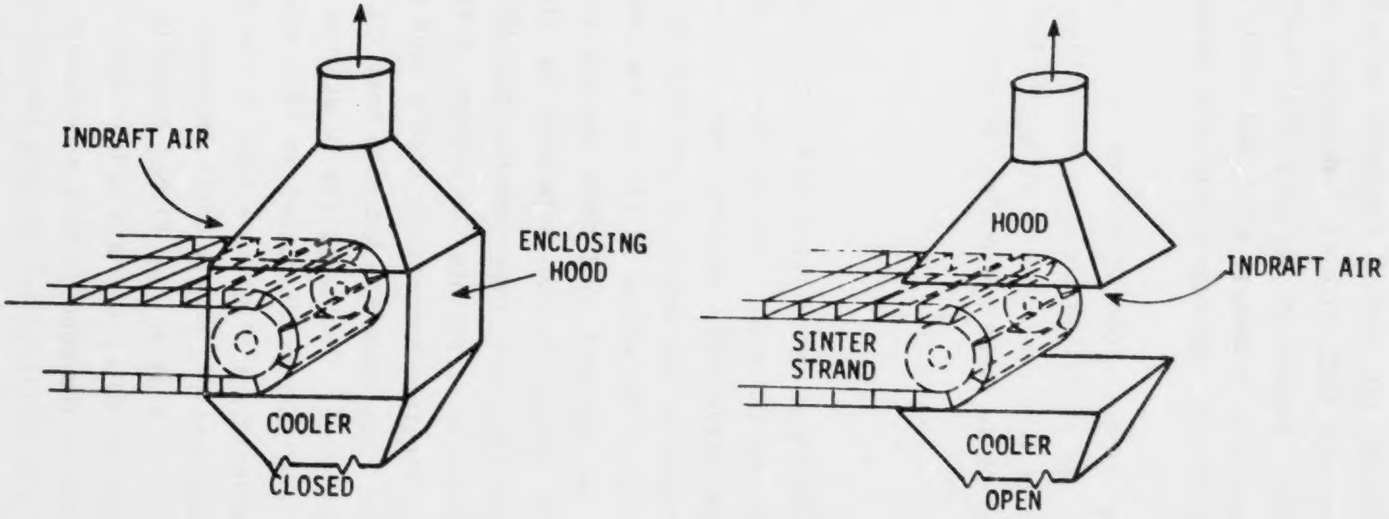

INDRAFT AIR
ENCLOSING HOOD
COOLER
CLOSED
HOOD
INDRAFT AIR
SINTER STRAND
COOLER
OPEN

Figure 2. Example of open and closed hooding on the discharge end of a sinter strand.[1]

Courtesy: Industrial Ventilation. 16th Ed. Published by American Conference of Governmental Industrial Hygienists. 1980.

than the lower explosive limit (LEL). In the electric arc furnace arrangement just mentioned, for example, a gap is provided to allow sufficient infiltration of air for combustion of carbon monoxide and dilution of its concentration (CO). This gap also permits the furnace to be tilted as illustrated in Figure 3.

In other arrangements, the hood is separated from the process for access purposes or to allow outside air to mix with and cool, dilute, or combust the process exhaust. The much larger open area of these arrangements requires much greater total exhaust flow. Two basic designs are used in the control of emissions from basic oxygen furnaces (BOF's), the so-called open-hood and the more-energy-efficient closed-hood approach. Application of a ventilation system to a BOF is more complex than most applications because the exhaust gas is primarily carbon monoxide, which, in open-hood systems, is combusted in the hood by the indraft air. This combustion leads to temperature and volume variations that must be accounted for in the design of the exhaust system. With closed-hood systems, the CO is not burned in the hood. The CO-rich gas is cleaned and then flared or used as a fuel. Thus, for closed-hood systems, air infiltration must be minimized to avoid explosions in the gas-cleaning system; its major advantage is the greatly reduced size of the gas cleaning equipment. Table 1 illustrates the difference in flow rate associated with these BOF systems. The difference in flow rate also dictates the type of control device used. A scrubber, for example, would be very expensive to operate on the high flows resulting from the open-hood approach.

Another common problem in many systems is the variability of conditions. Conditions may vary from one season to another (i.e., ambient temperature and/or ventilation requirements), from one heat to another, or even from one moment to the next as process conditions change. For example, Figure 4 illustrates the variation in gas flow during a heat in the BOF process. This variation occurs over a period of 15 to 20 minutes.

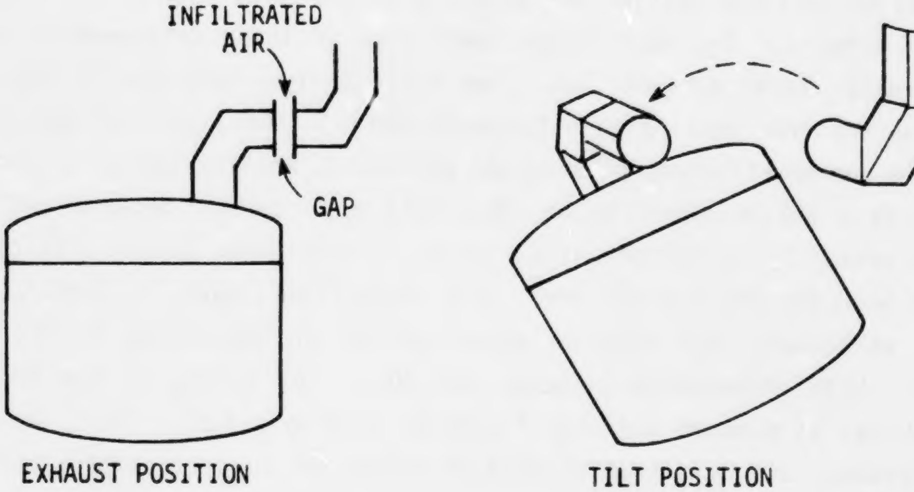

INFILTRATED AIR
GAP
EXHAUST POSITION
TILT POSITION

Figure 3. Direct shell evacuation on an electric arc furnace.

TABLE 1. COMPARISON OF PRINCIPAL DATA FOR 10, 30, AND 100 PERCENT COMBUSTION IN BOF HOODING[a]

Varying parameters	Type of hooding and combustion rate			
	Semi-open 100%	Semi-closed 30%	Closed 10%	Open 200%
Total gas volume, scfm	158,800	87,000	66,700	318,600
Theoretical gas temperature inlet hood, °F	4352	3992	3272	b
Heat to be removed in hood, 10^6 Btu/h	889	325	167	b
Fan horsepower of high-energy scrubber, kW	4100	2200	1640	8200

[a] Reference 8.

[b] NA = Not available.

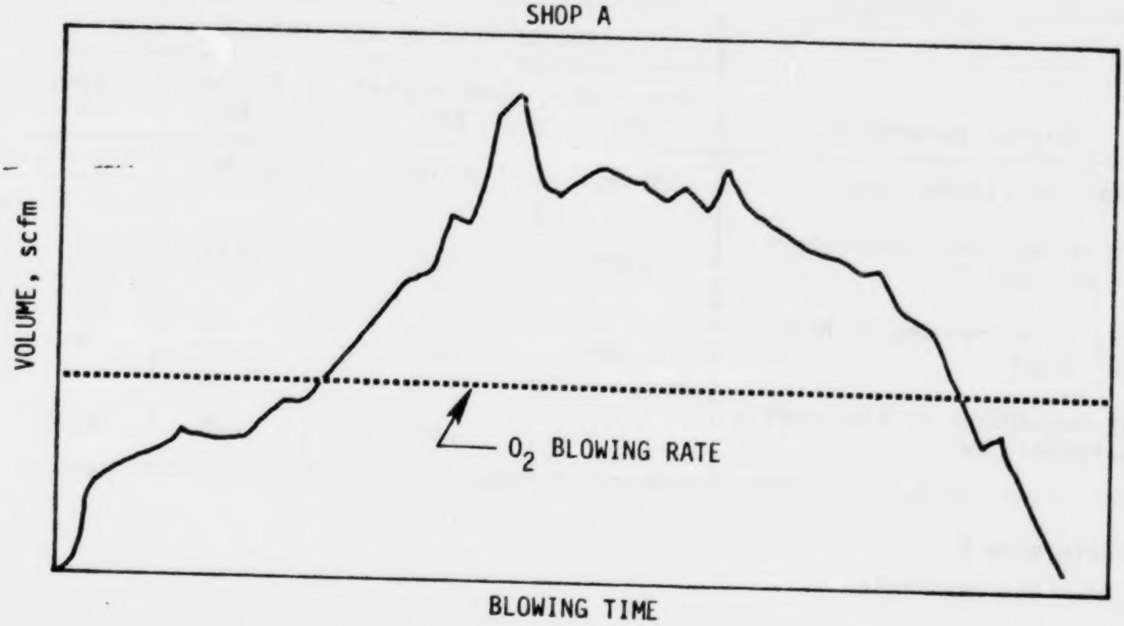


Figure 4. Variation in gas flow rate from a BOF during the course of a heat.

Courtesy: JAPCA, 18(2):98-101, February 1968. Article by D. H. Wheeler entitled "Fume Control in L-D Plants."

SECTION 3

HOOD DESIGN CONSIDERATIONS

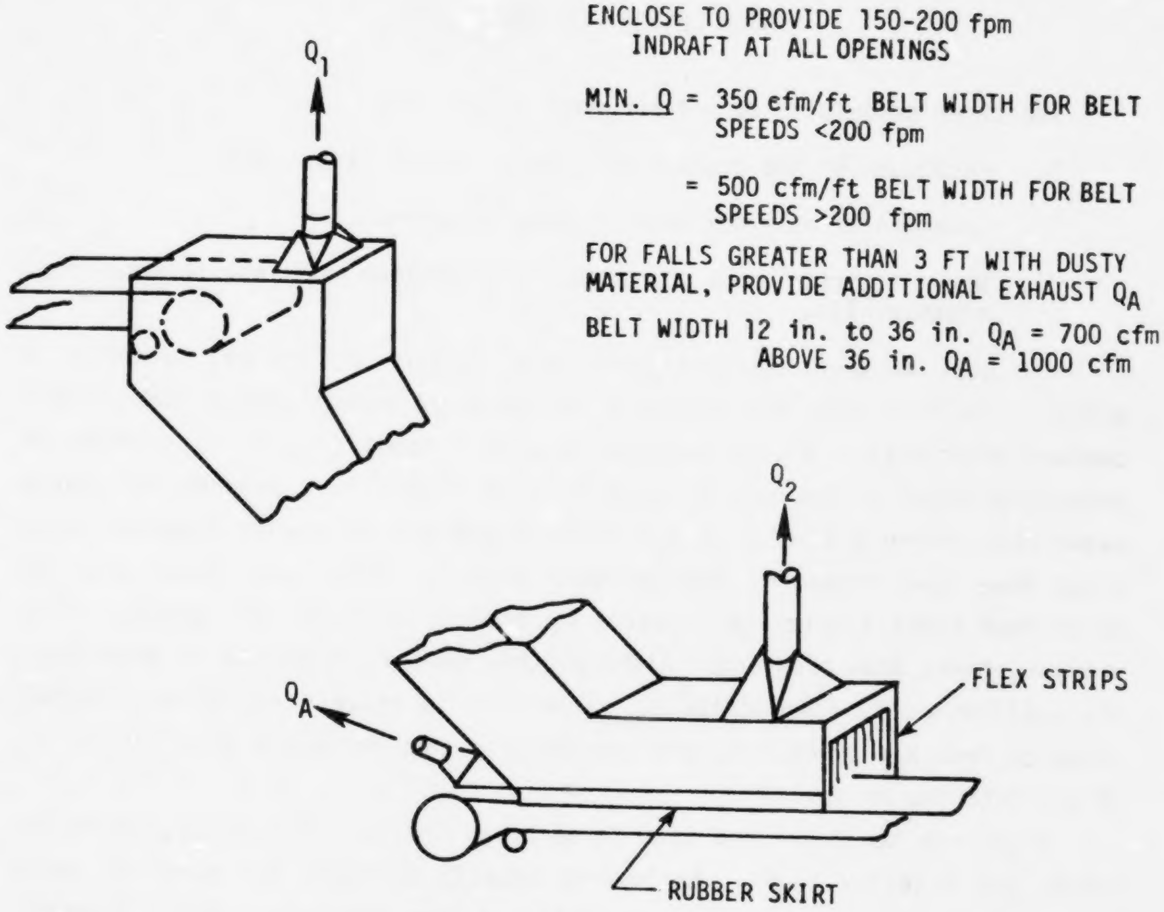
The three principles of optimum hood design are:

- Enclosure of the process or source insofar as possible.
- Location of exterior hood in path of exhaust.
- When exterior hood is used, minimization of interference from cross-drafts.

The goal of good hood design is high capture efficiency. Ideally, a process should be entirely enclosed, which would permit almost 100 percent capture efficiency. Simple conveyor transfer hoods (Figure 5) provide an example of total enclosure. Because frequent access to a process (to charge materials, remove products, or perform maintenance) is usually required, most hoods have open areas to provide this access. These open areas must be maintained under a negative pressure by drawing air into the system, which prevents fumes from escaping. Although this concept is simple in principle, its application is complicated by variations in process emissions, thermal currents from hot processes, and cross-drafts that interfere with the inflow of air into the hood.

Hoods can be classified into three broad groups: enclosures, receiving hoods, and exterior hoods. Enclosures usually surround the point of emission, but sometimes one face is partially or even completely open. Examples of enclosures are paint spray booths, abrasive blasting cabinets, totally enclosed bucket elevators, and enclosures for conveyor belt transfer points, screens, crushers, etc. The sides of the enclosure effectively reduce cross-drafts and also direct the plume toward the capture hood.

Receiving hoods are those in which the air contaminants are injected into the hoods and inertial forces carry these emissions into the hood. These hoods are generally applied to smaller processes that impart a velocity



ENCLOSE TO PROVIDE 150-200 fpm
INDRAFT AT ALL OPENINGS

MIN. Q = 350 cfm/ft BELT WIDTH FOR BELT SPEEDS <200 fpm

= 500 cfm/ft BELT WIDTH FOR BELT SPEEDS >200 fpm

FOR FALLS GREATER THAN 3 FT WITH DUSTY MATERIAL, PROVIDE ADDITIONAL EXHAUST Q_A

BELT WIDTH 12 in. to 36 in. Q_A = 700 cfm
ABOVE 36 in. Q_A = 1000 cfm

Figure 5. Conveyor transfer point hooding using total enclosure.

to the emissions, such as grinders and paint sprayers. They are not applicable to the large systems discussed in this manual.

Exterior hoods must capture air contaminants that are being generated from a point outside the hood itself. These hoods are generally used for large systems that generate heat and require frequent access. Figure 6 is a simplified diagram of an exterior or canopy-type hood. In this example, the hood design is augmented with baffles and dampers to direct suction to one of three sections, depending on the source of the emissions. This enhances capture efficiency by decreasing the effective face area.

The total air flow into the hood system is determined by Equation 1:

$$Q = A\ V \qquad \text{(Eq. 1)}$$

where Q = Total air flow, cfm
A = Cross-sectional area, ft^2
V = Air velocity perpendicular to open face area, ft/min

This simple equation is the root of inadequate system design. Because cost is directly proportional to flow, Q, the user is continually tempted to decrease either hood area, A, or face velocity, V. As described later, a decrease in either of these causes rapid deterioration in hood capture performance.

The desired air velocity, or capture velocity, designed into the hood must be based on experience, but the guidelines in Table 2 may be helpful. In many larger industrial processes, the third category--active generation into a zone of rapid air motion--is encountered, and face velocities in the range of 200 to 500 ft/min are required. Air motion or currents in the room may be caused by thermal drafts from hot processes, building drafts, movement of machinery or material, movement of the process, or rapid discharge of gaseous emissions.

Because all of these factors cannot be accurately evaluated, a high-efficiency hood must have high face velocities in which a large safety factor is incorporated. In addition, reducing cross drafts by using partial enclosures (both fixed and movable) will greatly enhance capture efficiency.

In the design of a hood system, it is useful to consider the concept of a null point. This point is defined as the point where the inertial energy (mass times velocity) of the emission has decreased to zero or been nullified. Because the mass of most emissions (gases and/or particles) is

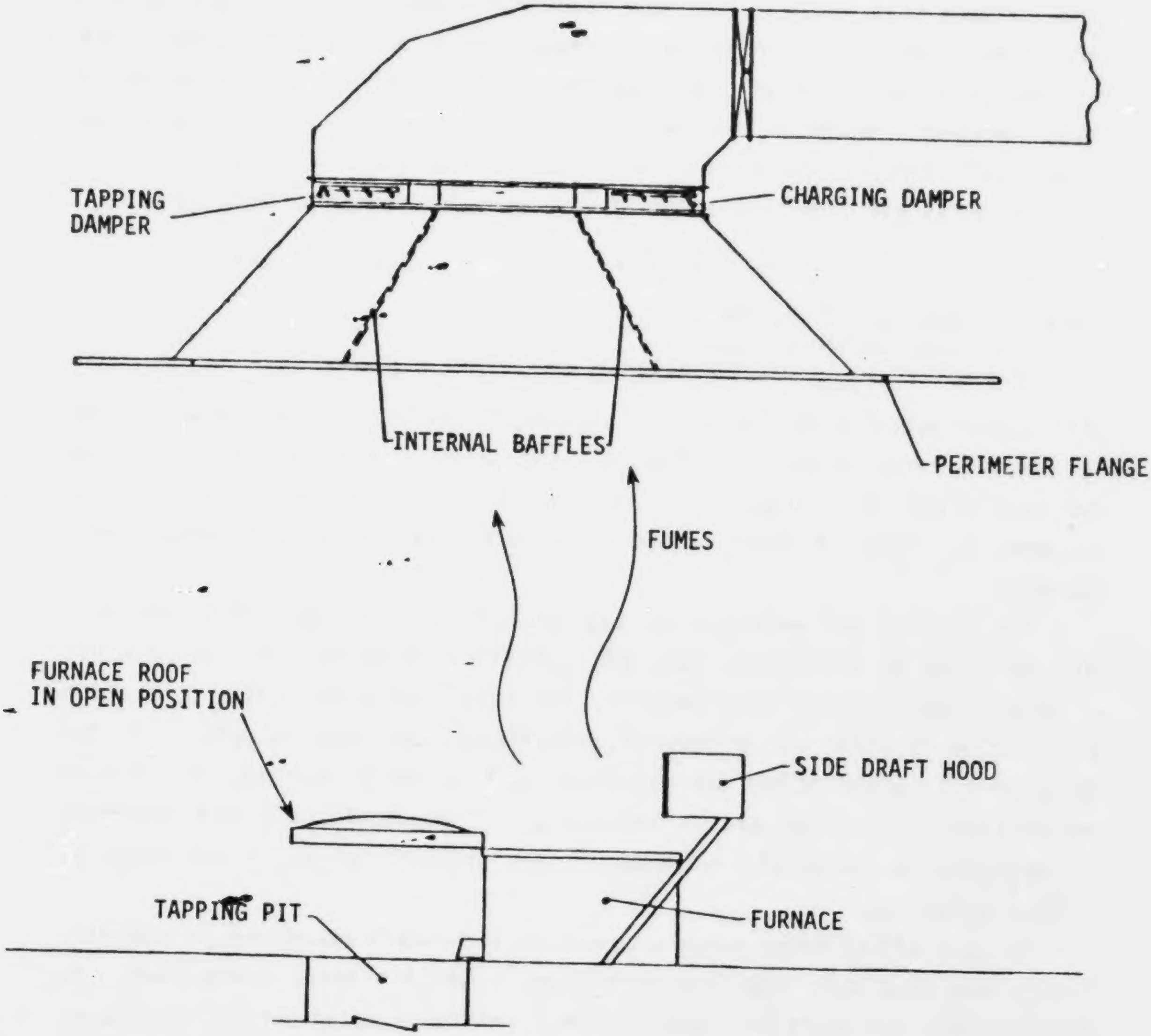

TAPPING DAMPER
CHARGING DAMPER
INTERNAL BAFFLES
PERIMETER FLANGE
FUMES
FURNACE ROOF IN OPEN POSITION
SIDE DRAFT HOOD
TAPPING PIT
FURNACE

Figure 6. Exterior EAF canopy hood.

TABLE 2. RANGE OF CAPTURE VELOCITIES

Condition of dispersion of contaminant	Examples	Capture velocity, ft/min
Released with practically no initial velocity into quiet air.	Evaporation from tanks; degreasing, etc.	50-100
Released at low velocity into moderately still air.	Spray booths; intermittent container filling; low speed conveyor transfers; welding; plating; pickling	100-200
Active generation into zone of rapid air motion.	Spray painting in shallow booths; barrel filling; conveyor loading; crushers; melting and refining	200-500
Released at high initial velocity into zone of very rapid air motion.	Grinding; abrasive blasting; tumbling	500-2000

NOTE: In each category above, a range of capture velocity is shown. The proper choice of values depends on several factors:

Lower end of range	Upper end of range
1. Room air currents minimal or favorable to capture.	1. Disturbing room air and thermal currents.
2. Contaminants of low toxicity or of nuisance value only.	2. Contaminants of high toxicity.
3. Intermittent, low production.	3. High production, heavy use.
4. Large hood, large air mass in motion.	4. Small hood, local control only.

Courtesy: Brandt, A. D. Industrial Health Engineering, John Wiley and Sons, New York. 1947.
Kane, J. M. Design of Exhaust Systems. Heating and Ventilating, 42, 68. November 1945.

smal their momentum is soon dissipated by air resistance. Hot process exhausts often have significant momentum due to thermal updraft. Examples are electric arc furnace emissions and coke pushing emissions. Thermal momentum can be misinterpreted in the sense that one might think the gases would be easily captured because they are headed directly into the capture hood. If the upward velocity is greater than the hood face capture velocity, however, the gas stream will be deflected to the side as if it had struck a barrier. This is illustrated in Figure 7.

At the null point the emissions have no momentum of their own, and if an adequate draft or air velocity toward the hood is provided at the null point, the contaminants will be captured. What constitutes an adequate velocity toward the hood depends on the drafts in the area, and therefore cannot be determined precisely.

Establishing the null point in advance for a new process is not always possible. For existing equipment, however, direct observation will usually establish a locus of null points. In the absence of external disturbances, any positive velocity toward the hood at the null point will give complete capture. In practice, however, complete capture is difficult to achieve because of drafts and thermal currents that disturb the air flow and prevent the formation of the null point. Sufficient velocity must be induced to overcome the disturbance caused by drafts and thermal currents. Because these drafts and thermal currents vary with the activity near the process, an exact entrainment velocity cannot be calculated; therefore, a safety factor must be incorporated to ensure good capture.

3.1 DETERMINING AIR FLOW REQUIREMENTS

3.1.1 Cold Processes

As shown in Figure 8, air moves from all directions toward openings under suction.[9] By definition, flow contours are lines of constant velocity in front of a hood. Similarly, streamlines are lines perpendicular to velocity contours.

The equation for air flow around free-hanging round hoods and rectangular hoods that are approximately square is[10]:

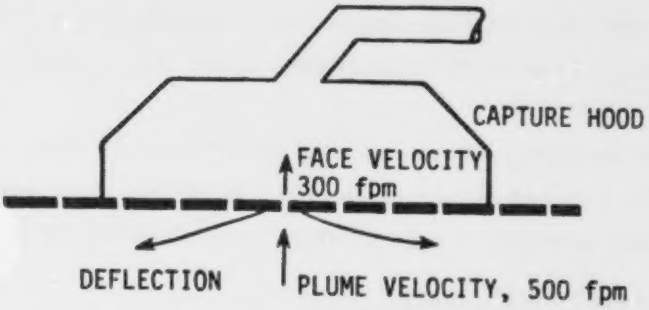

CAPTURE HOOD
FACE VELOCITY
300 fpm
DEFLECTION
PLUME VELOCITY, 500 fpm

Figure 7. Effect of excessive plume velocity.

$$V = \frac{Q}{10X^2 + A} \quad \text{(Eq. 2)}$$

where V = Centerline velocity at X distance from hood, ft/min
X = Distance outward along axis, ft (Equation is accurate only for limited distance of X, where X is within 1.5D.)
Q = Air flow, cfm
A = Area of hood opening, in ft²
D = Diameter of round hoods or side of essentially square hoods

As shown in Equation 2 and Figure 8, velocity decreases rapidly with increasing distances from the hood and varies almost inversely with the square of the distance. The velocity decreases less rapidly with a flanged hood, as shown in Figure 9.

Where distances of X are greater than 1.5D (as is the case in most applications), the velocity decreases less rapidly with distance than Equation 2 indicates. Figure 10 illustrates other hood types and gives the air volume formulae that apply.

In addition to canopy-type hooding systems, many other configurations of hood systems are applied to spray booths, grinding, and open tanks. Few of these systems have exhaust flows greater than 100,000 cfm.

3.1.2 Hot Processes--High Canopy Hoods

In hot processes, significant quantities of heat are transferred to the surrounding air by conduction and convection, and a thermal draft is created that causes a rising air current. The design of the hood and the ventilation rate provided must take this thermal draft into consideration.

As the heated air stream that rises from a hot surface moves upward, it mixes turbulently with the surrounding air. The higher the air column rises, the larger it becomes and the more it is diluted with ambient air. As illustrated in Figure 11, the rising air column expands approximately according to the following empirical formula:

$$D_c = 0.5\, x_f^{0.88} \quad \text{(Eq. 3)}$$

where D_c = The diameter of the hot column of air at the level of the hood face, ft

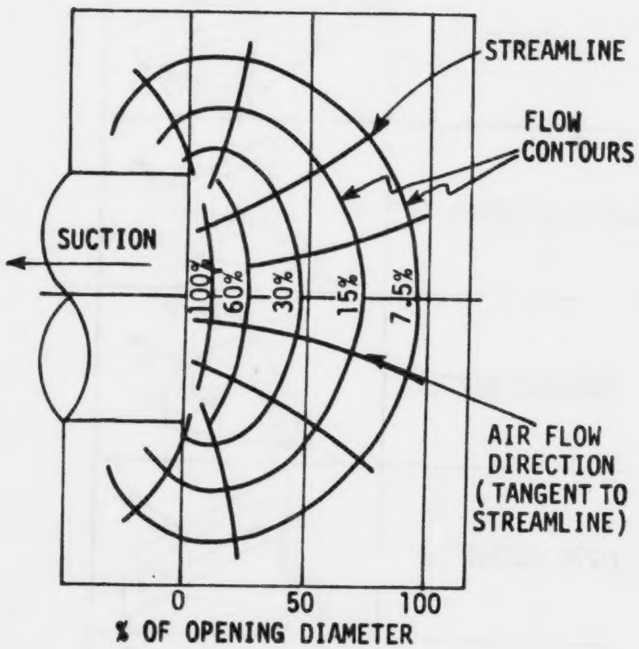

STREAMLINE
FLOW CONTOURS
SUCTION
100%
60%
30%
15%
7.5%
AIR FLOW DIRECTION (TANGENT TO STREAMLINE)
0
50
100
% OF OPENING DIAMETER

Figure 8. Velocity contours (expressed in percentage of opening velocity) and streamlines for circular openings.

Courtesy: Silverman, L. Velocity Characteristics of Narrow Exhaust Slots. Journal of Industrial Hygiene and Toxicology, 24, 267. November 1942.

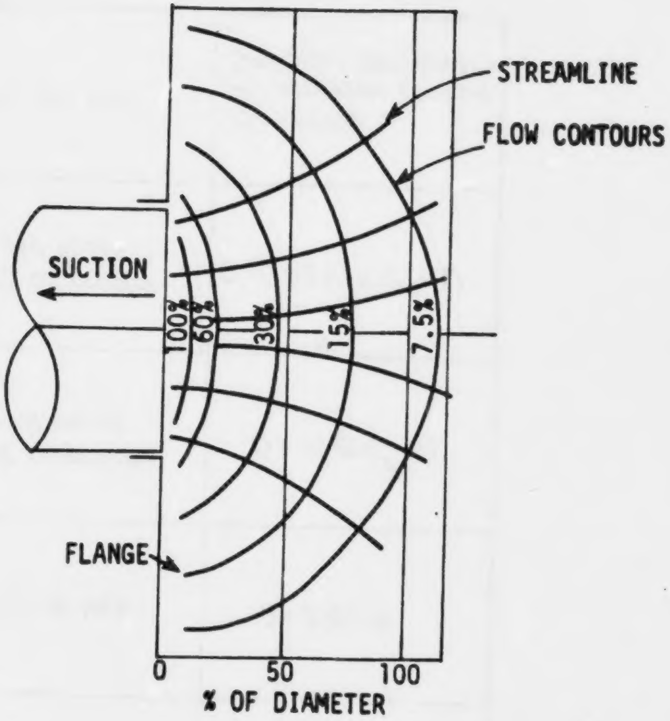

STREAMLINE
FLOW CONTOURS
SUCTION
100%
60%
30%
15%
7.5%
FLANGE
0
50
100
% OF DIAMETER

Figure 9. Velocity contours and streamlines for flanged hood.

Courtesy: Silverman, L. Centerline Velocity Characteristics of Round Openings Under Suction. Journal of Industrial Hygiene and Toxicology, 24, 259. November 1942.

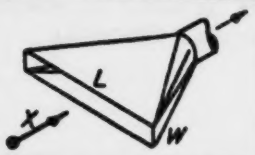
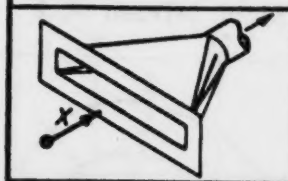
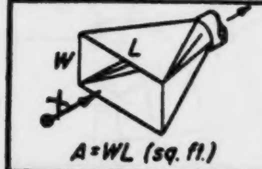
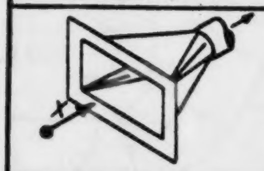
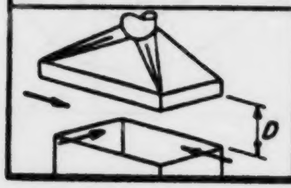
HOOD TYPE	DESCRIPTION	ASPECT RATIO, $\frac{W}{L}$	AIR VOLUME
	SLOT	0.2 or less	$Q = 3.7\,LVX$
	FLANGED SLOT	0.2 or less	$Q = 2.8\,LVX$
	PLAIN OPENING	0.2 or greater and round	$Q = V(10X^2 + A)$
	FLANGED OPENING	0.2 or greater and round	$Q = 0.75V(10X^2 + A)$
	CANOPY	To suit work	$Q = 1.4PDV$ P = perimeter of work D = height above work

Figure 10. Formulas for estimating hood air flows.

Courtesy: DallaValle, J. M. Exhaust Hoods. Industrial Press, New York. 1946.
Silverman, L. Velocity Characteristics of Narrow Exhaust Slots. Journal of Industrial Hygiene and Toxicology, 24, 267. November 1942.

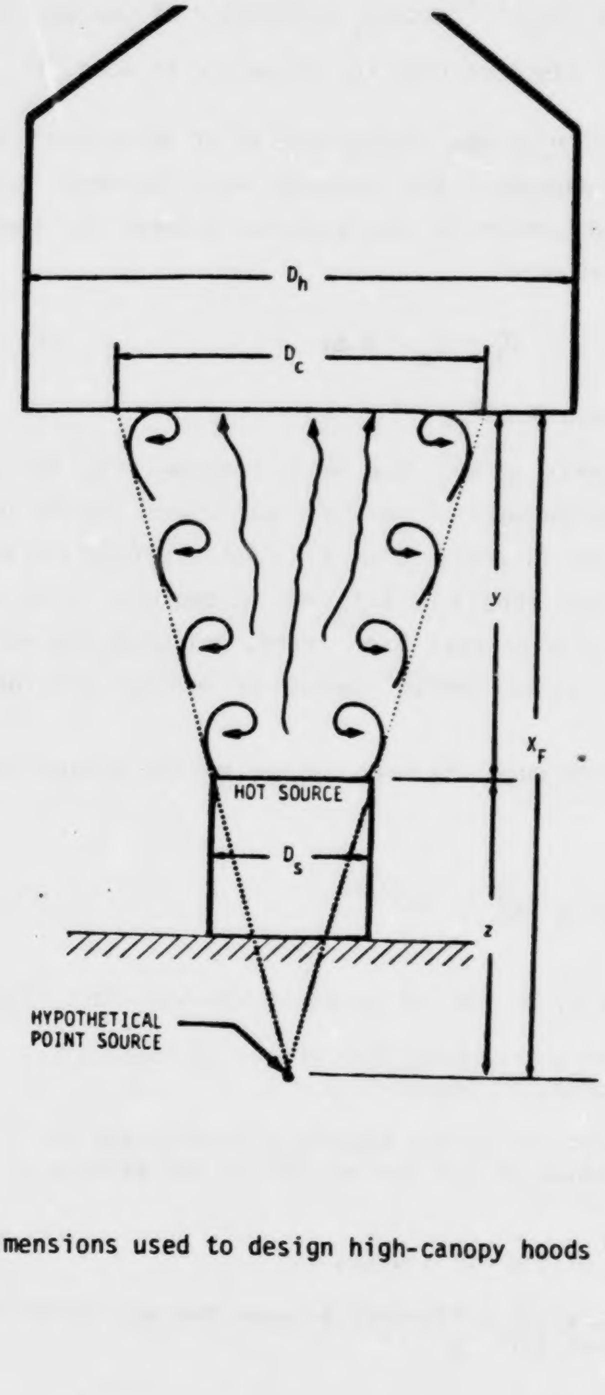

D_h
D_c
y
X_F
HOT SOURCE
D_s
z
HYPOTHETICAL POINT SOURCE

Figure 11. Dimensions used to design high-canopy hoods for hot sources.[12]

x_f = The distance from the hypothetical point source to the hood face, ft (equal to y + z)

where $z = (2D_s)^{1.38}$ and D_s = diameter of source, ft

y = Distance from top of source to hood, ft

To allow for drift in the rising column of emissions caused by crossdrafts and air disturbances, the designer must increase the overall hood diameter by adding 80 percent of the distance between the hood and the process, as shown in Equation 4.

$$D_h = D_c + 0.8y \qquad \text{(Eq. 4)}$$

where D_h = Overall hood diameter, ft

Where cross drafts occur, the hood diameter may be increased still farther and the distance between the hood and source may be decreased. When possible, side shields in the form of steel sheets (curtain walls) or chains suspended from the hood should be utilized to decrease cross drafts. Asbestos and tarpaulin curtains have been tried, but they are rarely successful because their light weight makes them tear easily and they do not hang straight.

The total flow through the hood system may be estimated by the use of Equation 5:

$$V_f = \frac{37}{(x_f)^{0.29}} \left(\frac{q}{60} A_s \ \Delta t\right)^{0.33} \qquad \text{(Eq. 5)}$$

where V_f = The velocity of the hot gases at the hood face, ft/min

x_f = The height of the hood face above the theoretical point source, ft (equal to y + z)

q = The natural convection heat loss coefficient (a nominal value of 0.4 can be used in the absence of other data[11])

A_s = The area of the hot source, ft²

Δt = The temperature difference between the hot source and the ambient air, °F

In addition to the volume of the hot gases rising through the hood area defined by diameter D_c (Figure 3-8), room air is also drawn into the hood through the balance of the hood area. Estimates of the desired velocity of the air through this portion of the hood depend entirely on engineering judgment and are based on expected cross-drafts, air disturbances, and the toxicity of the emissions.

A velocity between 100 and 200 ft/min is recommended, and the velocity should increase with greater air disturbances. In extreme cases in which more violent reactions occur with sudden heat release (such as in the charging and tapping of steelmaking furnaces), even greater velocities are required:

$$Q = V_f A_c + V_r (A_h - A_c) \qquad \text{(Eq. 6)}$$

where Q = Total hood flow, cfm

V_f = Velocity of hot air, ft/min

A_c = Area of hood face through which hot gases enter ($= \pi D_c^2/4$), ft²

V_r = Desired velocity of air entering balance of hood (100 to 200 ft/min)

A_h = Area of total hood, ft²

The control of emissions from sources that are other than circular in shape is best handled by hoods of the appropriate shape. Thus, a rectangular source would require a rectangular hood to minimize the ventilation requirements. The equations used for circular hoods are appropriate for rectangular hoods, but increases of 0.8 times the distance to the source (y) should be made in both length and width.

Total hood volume determines retention time in the hood. For intermittent processes of short duration, such as charging, a large hood has the advantage of containing the exhaust gas for several minutes until the ventilation system can withdraw the fumes. Partitions can be added that will not only minimize cross-drafts, but also essentially increase hood size and retention time. In any event, it is difficult to predict performance based solely on theoretical design. Scale model studies can be helpful during the design stage, but final modifications in the field may be necessary based on observation of system performance.

3.1.3 Hot Processes--Low Canopy Hoods

An exact distinction cannot be made between a low and high canopy hood, but a low hood is usually defined as one in which the distance between the hood and the source does not exceed the diameter of the source, or 3 feet, whichever is smaller. The primary difference in the design of low hoods is that the hood diameter and source diameter are essentially the same. A safety factor is usually included, and for practical purposes, the hood diameter should exceed the source diameter by at least one foot. The dimensions of larger rectangular hoods should exceed the source's dimensions by at least one foot in all directions.

For circular hoods, the total exhaust flow may be determined by the following equation[12]:

$$Q = 4.7(D_h)^{2.33}(\Delta t)^{0.417} \qquad \text{(Eq. 7)}$$

where D_h = Hood diameter in feet and is equal to the source diameter plus 1 or 2 feet

For rectangular hoods, the exhaust flow may be determined by:

$$Q = (6.2)W^{1.33}\,(\Delta t)^{0.417}L \qquad \text{(Eq. 8)}$$

where W = Hood width, which is 1 to 2 feet larger than the source width

L = Length, which is 1 to 2 feet longer than the source length

Lowest flow rates are achieved with close-fitting hoods. Figure 12 illustrates the latest design for hooding a hot metal transfer operation. Note that the open area is essentially limited to the hood slot through which the metal is poured.

3.1.4 Building Evacuation

A building acts as a large process enclosure. By drawing air through the building and out the roof, the building essentially serves as a hood.

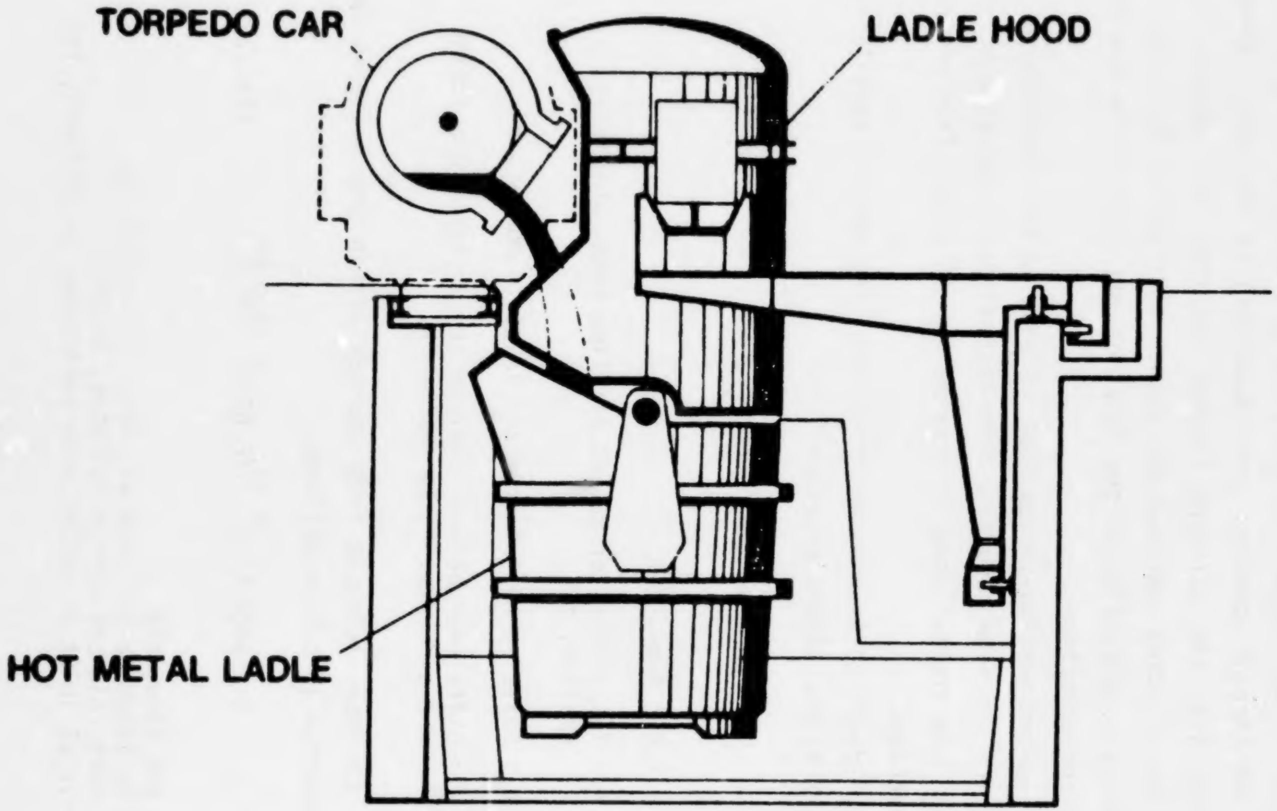

TORPEDO CAR
LADLE HOOD
HOT METAL LADLE

Figure 12. Schematic arrangement of ladle hood for reladling emission control. Courtesy: Pennsylvania Engineering Corporation, Pittsburgh.

The large size of a building compared with a more closely fitted hood requires utilization of a much larger exhaust flow. Many large buildings, and especially those that contain processes that release heat, are ventilated by natural draft. Heat released by the processes warms the air, which rises and is pushed out through openings (roof monitors) in the roof. Cooler ambient air is drawn into the building through openings near ground level. Wind action at the building openings may either increase or decrease the natural ventilation rate, depending on the location and size of the openings and the wind speed and direction.

A forced-ventilation system that is applied to a building must be sized to include the air flow resulting from natural draft and also maintain sufficient draft into the building to prevent any emissions from escaping through building openings.

For buildings containing hot processes, the natural ventilation rate can be estimated by the ASHRAE equation[11]:

$$Q = 9.4(AL)^{0.5}(\Delta t_{avg})^{0.5} \qquad \text{(Eq. 9)}$$

where Q = Air flow, cfm

A = Total inlet or outlet air flow area, whichever is smaller, ft²

L = Building height from air inlet to outlet, ft

Δt_{avg} = Difference between average temperature in building and air entering building, °F

When the heat released from sources within the building can be quantified, Equation 10 can be utilized:

$$Q = 20(0.67L)^{0.33}(0.67H)^{0.33}A^{0.67} \qquad \text{(Eq. 10)}$$

where Q = Air flow, cfm
L = Building height from air inlet to outlet, ft
H = Heat released within building, Btu/min
A = Total inlet or outlet area (whichever is smaller), ft²

Building ventilation requirements may also be defined in terms of air changes per unit of time. Again, a great deal of judgment enters into the selection of the number of changes required, which depends on the heat generated in the building and the industrial hygiene considerations regarding fumes and gases within the building. On the order of at least 20 air changes per hour are required for metallurgical processes.

The distribution of ventilation is also very important. Uncontrolled air flowing into a building as a result of negative pressure in the building or because of poorly designed air-supply distributors not only may cause recirculation of contaminants, but also may upset the local ventilation systems. Therefore, the amount of air, the location of its entry into the building, and its direction must be controlled. Figure 13 shows a controlled air supply that results in a convective flow from a heat source (such as a ladle of molten metal) rising to be exhausted through a roof ventilator. Figure 14 shows an uncontrolled air supply, which results in a disrupted plume and recirculation of the contaminant throughout the building. The latter could cause a buildup of contaminants in the building and possible leakage to the outside air.

Because of the huge air volumes it would require (on the order of 5 to 10 million cfm), true building evacuation is rare. Care must be taken in closing the roof monitors or ventilators in a building and replacing this natural ventilation with induced-draft ventilation. If the induced draft is inadequate, both ambient dust and heat levels in the building can rise rapidly and create health and safety hazards. This is particularly true in hot climates.

3.1.5 Push-Pull Systems

Hood capture efficiency can sometimes be improved by the use of a push-pull or air curtain approach. This approach involves a source of compressed air (push) to direct the emission plume toward the exhaust hood (pull). These applications are used to control blast furnace casthouses, copper converters, and electric arc furnace enclosures. Figure 15 illustrates the general principle of the push-pull system. The effective face velocity of a

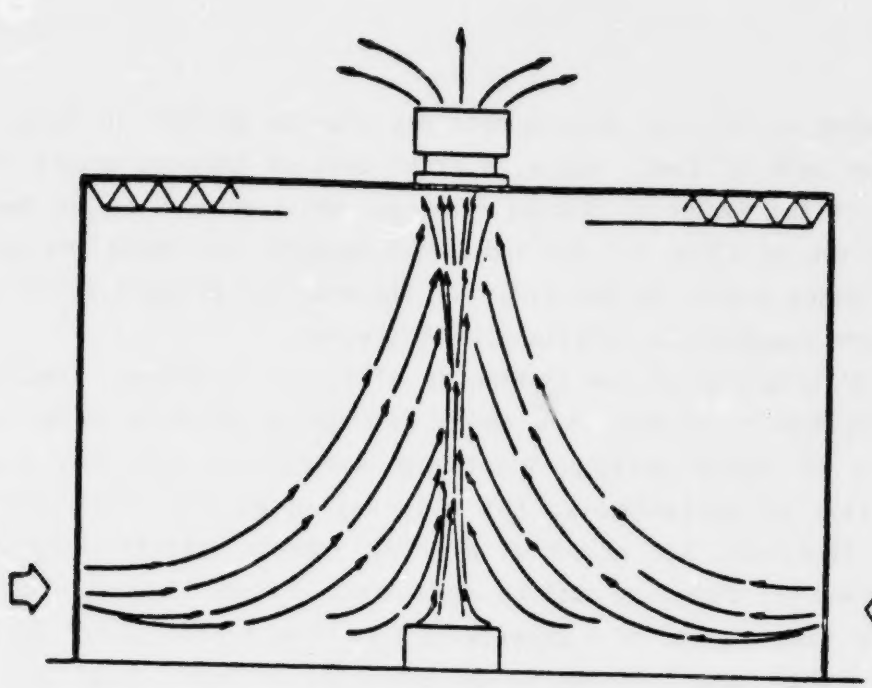


Figure 13. Controlled airflow from a heated source.[13]

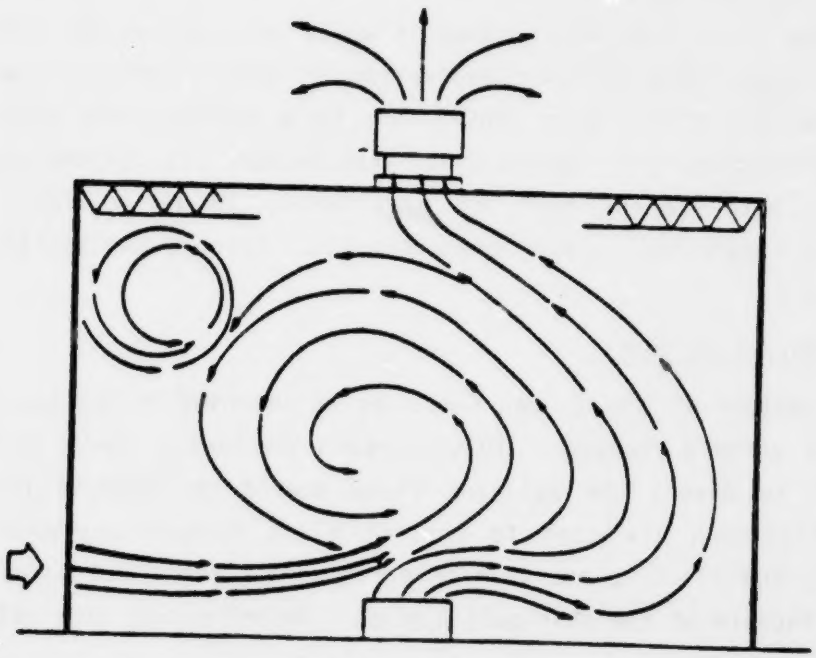


Figure 14. Uncontrolled airflow from a heated source.[13]

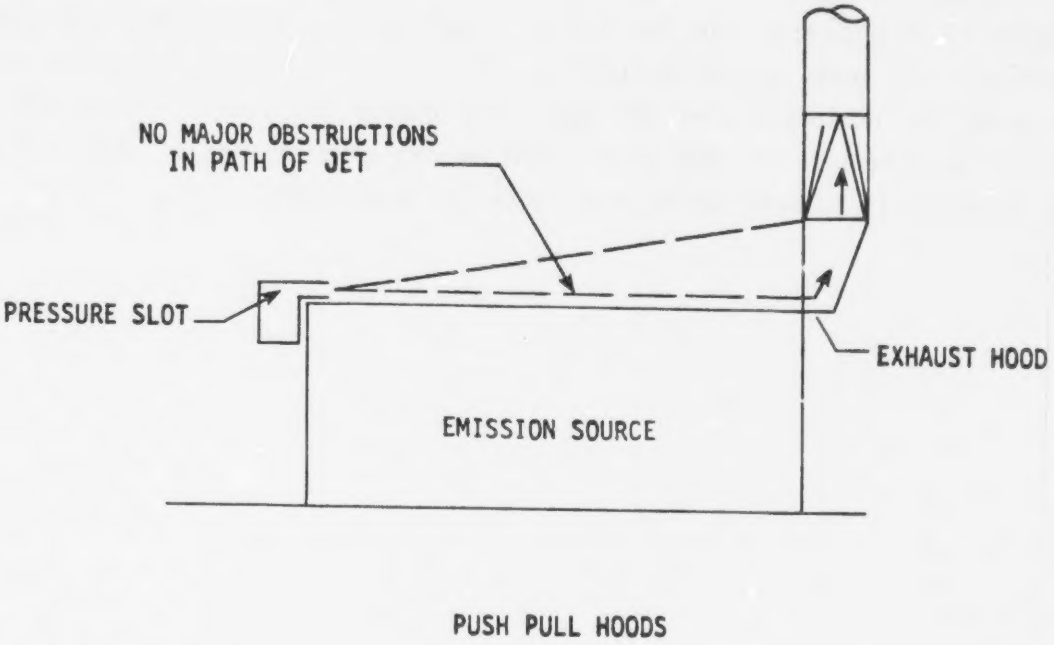

NO MAJOR OBSTRUCTIONS IN PATH OF JET
PRESSURE SLOT
EXHAUST HOOD
EMISSION SOURCE
PUSH PULL HOODS

Figure 15. General principle of the push-pull (air-curtain) type system.[9]

blowing source is sustained at much larger distances than that of a suction source (as illustrated in Figure 16).

In one system used to control tapping emissions from a blast furnace, an air curtain in front of the taphole directs the emissions into the capture hood above the taphole in an arrangement very similar to that in Figure 15. Figure 17 illustrates the air curtain application on a copper converter furnace. The total enclosure uses an air curtain to prevent emissions from escaping the enclosure when the doors are opened for access. Each application is unique in its design and must be evaluated by actual observation. It is difficult to base suitability on design data alone.

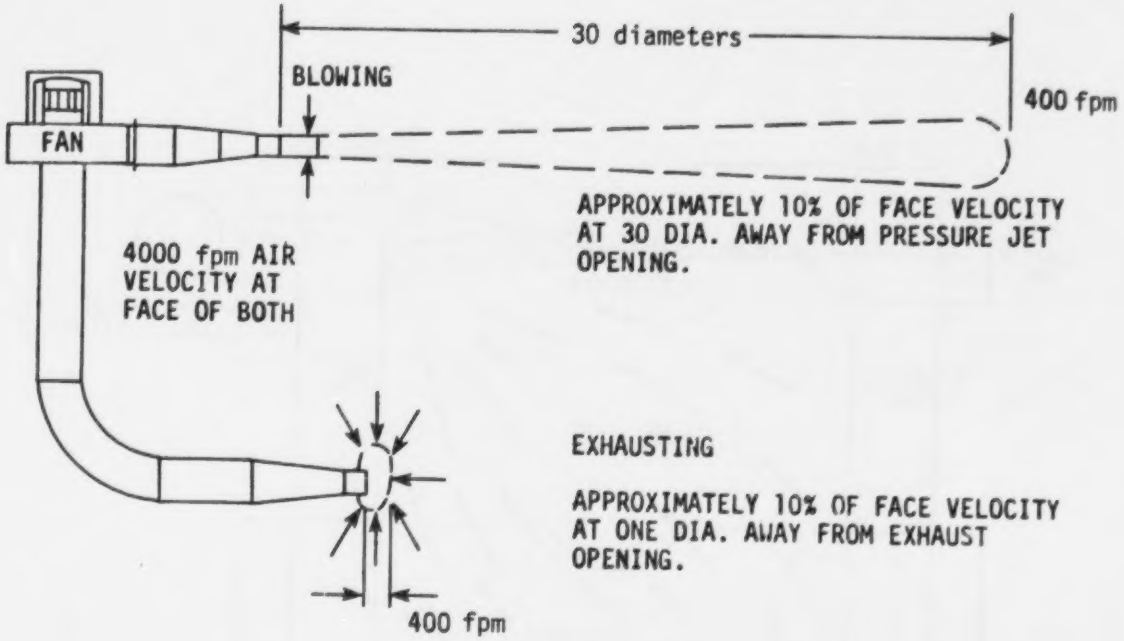

30 diameters
BLOWING
400 fpm
FAN
APPROXIMATELY 10% OF FACE VELOCITY AT 30 DIA. AWAY FROM PRESSURE JET OPENING.
4000 fpm AIR VELOCITY AT FACE OF BOTH
EXHAUSTING
APPROXIMATELY 10% OF FACE VELOCITY AT ONE DIA. AWAY FROM EXHAUST OPENING.
400 fpm

Figure 16. Comparison of face velocity decay for blowing versus exhausting.

Courtesy: Industrial Ventilation, 16th Ed.

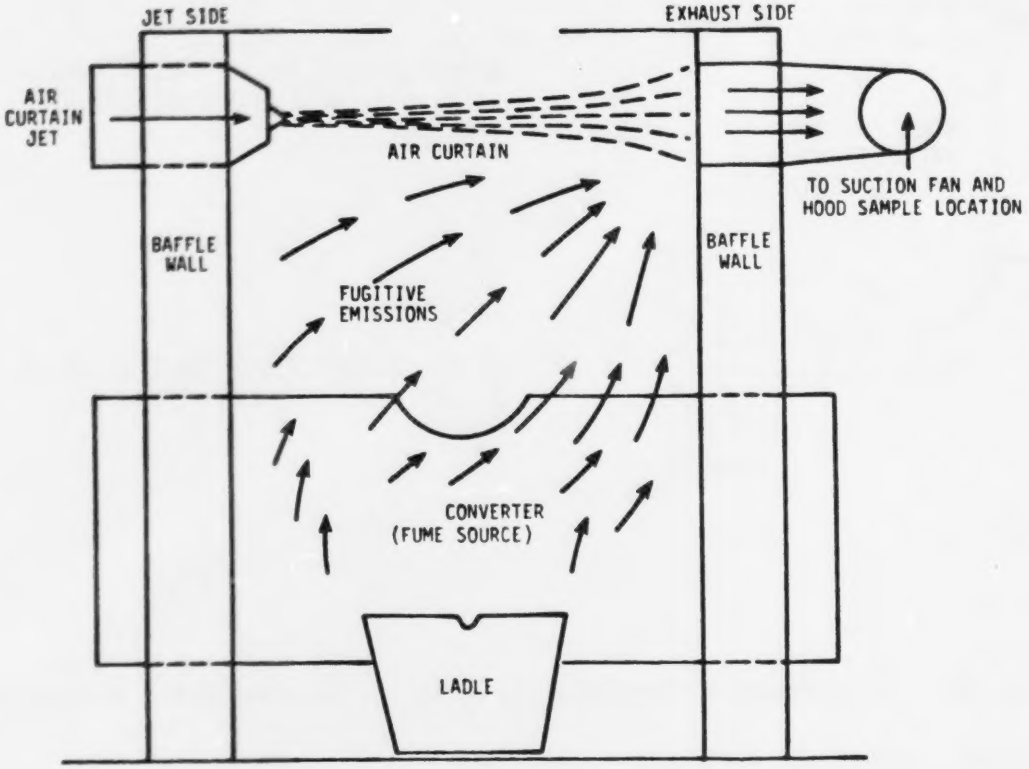

JET SIDE
EXHAUST SIDE
AIR CURTAIN JET
AIR CURTAIN
TO SUCTION FAN AND HOOD SAMPLE LOCATION
BAFFLE WALL
BAFFLE WALL
FUGITIVE EMISSIONS
CONVERTER (FUME SOURCE)
LADLE

Figure 17. Air curtain control system on a copper converter.

SECTION 4

DUCT DESIGN AND CONSIDERATIONS

The three design principles for ducting are:

- Minimization of changes in flow direction.
- Maintenance of smooth duct surfaces.
- Avoidance of abrupt expansions.

Basically, a duct is a pipe or channel that conveys a gas and contained emissions from a collection point to a more convenient point for rehandling, cleaning, or blending. In some cases a duct also acts as a cooler.

Duct configurations range from small-diameter ducting (6 to 12 in.) to ducting having cross-sectional areas of 600 square feet or more. Ductwork can be a combination of circular, square, or rectangular, based on location, space limitations, equipment or building design, and length of run to the control point. Flows should be continuous and smooth in direction, with no abrupt expansions or contractions. Larger duct sizes usually are fabricated in the field in square or rectangular cross sections. Circular ducts are limited in size by the availability of plate widths. The cost of wider plates must be balanced against forming and welding costs. Theoretically, a circular cross section is preferable because it minimizes friction losses and nonstreamline flow. Duct configuration is generally dictated by economics, however. Duct thickness in large metallurgical systems usually varies from 1/4 to 3/8 in. Strengthening ribs and expansion joints are generally required for larger sections (over about 6 feet in diameter or square). Expansion joints are required for ductwork carrying hot gases (above about 300°F). If temperature fluctuations occur regularly, cracking of the expansion joints and the resultant leakage can be a major source of maintenance problems.

Ducting must be designed to deal with the following gas conditions: temperature, abrasiveness, acidity, dust concentration, and moisture. For temperature control, the ductwork may be lined with refractory materials or water-cooled in its entirety. An example of this is the BOF shop, where the hood and duct above the furnace are cooled by a variety of designs that are discussed later.

Damage from abrasion, acidity, and moisture may be controlled by special types of refractory or, depending on the temperature, by the use of various coatings or special alloyed steels. Carbon steel, for example, generally is considered suitable for gas temperatures up to 1000°F. Above this temperature, stainless steel or refractory-lined carbon steel is required. Acid condensation (primarily from sulfates) is a problem if gas temperature falls below the acid dew point. In copper converters, sulfur trioxide (SO_3) typically constitutes about 1 percent of the sulfur dioxide (SO_2) present, or about 0.02 to 0.1 percent of the total gas stream. This can condense and form sulfuric acid. Figure 18 indicates the dewpoint of air containing various SO_3 concentrations.

4.1 TRANSPORT VELOCITIES

The minimum design duct velocity is that required to prevent buildup in a duct. At elbows and other sections where the gas stream slows down, provision should be made for inspection for dust dropout and for cleaning. If the substances in the gas tend to be sticky, or if moisture condensation is possible, cleanout ports or flanged sections should be provided to gain access. Buildup decreases the effective duct cross-sectional area and increases transport velocity; the latter counterbalances the former and thus prevents further buildup. If buildup is due to stickiness or moisture, however, it can proceed to the point of total pluggage, especially in smaller branch ducts. A further concern caused by buildup is the possibility of its breaking loose during startups or vibration. Serious buildup can also create an excessive mechanical load on the duct structure and supports.

Normally, the minimum transport velocity required is greater than that required merely to prevent settling or buildup. Buildup or pluggage can cause system upset in the main duct or in any of the branches, and this in

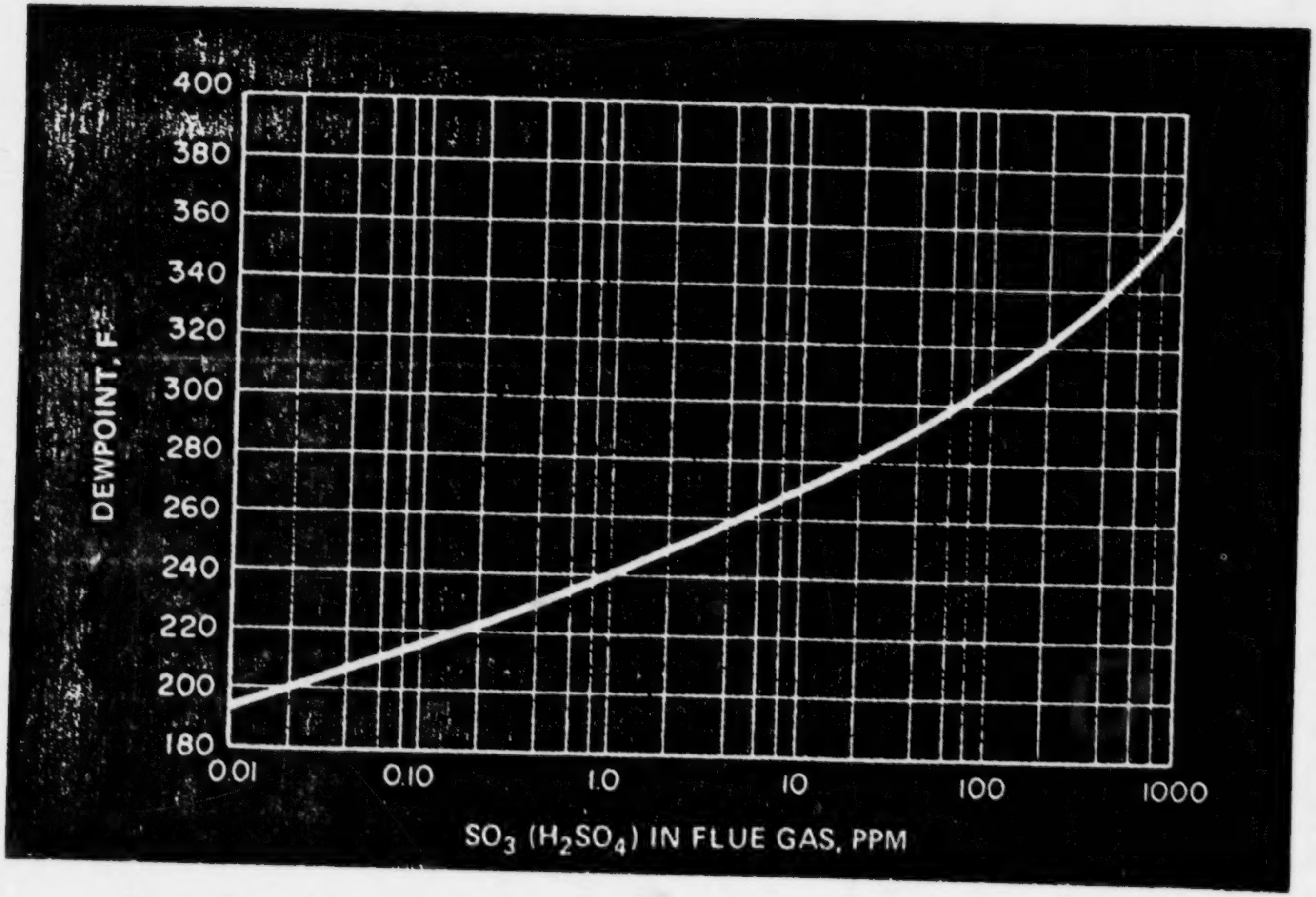


Figure 18. Dew-point of air containing various SO_3 concentrations.

Courtesy: "Combating Fuel Oil Heating Problems", Plant Engineering, January 7, 1974.

turn affects the remainder of the system because of the increase in resistance and decrease in flow in the blocked section. Anything that decreases the duct size (such as damage from outside forces) affects overall collection performance.

A change in the temperature of the gas at any point can also affect the overall collection efficiency of the system.

The transport velocity must account for the velocity needed for gas and particulate removal, resistance due to friction along the duct surface, and dynamic losses due to air turbulence.

Table 3 gives a recommended range of design velocities. Metallurgical process control systems with heavy dust loadings operate in the transport velocity range of 3500 to 4500 fpm. When the dust consists primarily of small particles at low concentration, lower transport velocities are possible. Higher velocities are still preferable, however, to decrease duct diameter. Duct cost savings normally outweigh the energy penalty of the higher velocity.

4.2 ENERGY LOSSES

Figure 19 summarizes the three pressure measurements of concern in a duct. The sum of the energy losses that must be considered in the duct system are:

- Inertia - The energy required to accelerate the gas from zero to duct velocity is equal to $(V_a/4005)^2$, where V_a is the gas velocity in feet per minute. This value is referred to as velocity pressure or as the velocity head (h_v). Table 4 can be used to convert velocity (V_a or V) in feet per minute to velocity pressure (h_v or VP) in inches of water.[2]

- Straight duct friction - Friction loss (or pressure drop) in straight duct runs is usually negligible relative to the pressure drop required for elbows, branch entries, and the control device. As shown in the following equation, the total loss in 500 feet of 10-foot diameter duct (assuming V = 4000 fpm) is about 0.5 in. H_2O. The equation used for clean round ducts is:

$$f = \frac{2.74 \left(\frac{V}{1000}\right)^{1.9}}{D^{1.22}} \qquad \text{(Eq. 11)}$$

TABLE 3. RANGE OF DESIGN VELOCITIES[a]

Nature of contaminant	Examples	Design velocity
Vapors, gases, smoke	All vapors, gases, and smokes	Any desired velocity (economic optimum velocity usually 1000-1200 fpm)
Fumes	Zinc and aluminum oxide fumes	1400-2000
Very fine light dust	Cotton lint, wood flour, litho powder	2000-2500
Dry dusts and powders	Fine rubber dust, Bakelite molding powder dust, jute lint, cotton dust, shavings (light), soap dust, leather shavings	2500-3500
Average industrial dust	Sawdust (heavy and wet), grinding dust, buffing lint (dry), wool jute dust (shaker waste), coffee beans, shoe dust, granite dust, silica flour, general material handling, brick cutting, clay dust, foundry (general), limestone dust, packaging and weighing, asbestos dust in textile industries	3500-4000
Heavy dusts	Metal turnings, foundry tumbling barrels and shakeout, sand blast dust, wood blocks, hog waste, brass turnings, cast iron boring dust, lead dust	4000-4500
Heavy or moist	Lead dust with small chips, moist cement dust, asbestos chunks from transit pipe cutting machines, buffing lint (sticky), quick-lime dust	4500 and up

A rule of thumb for items or contaminants not tested is to operate in the 3500-4500 fpm transport velocity range.

[a] From Reference 14.

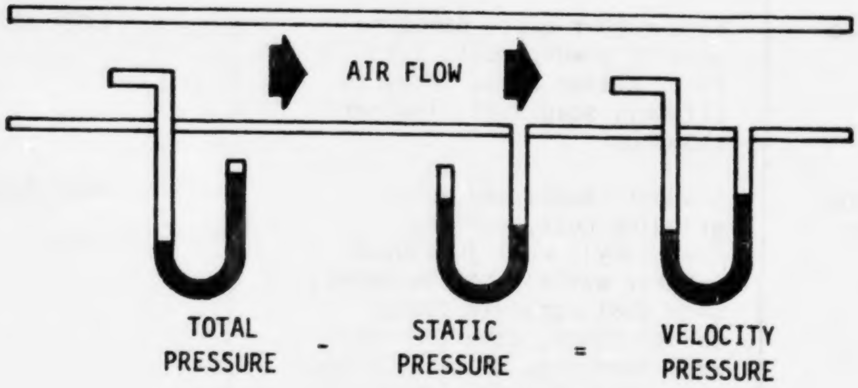

AIR FLOW
TOTAL PRESSURE
-
STATIC PRESSURE
=
VELOCITY PRESSURE

Figure 19. Pressure measurements in ducts.[14]

TABLE 4. CONVERSION TABLE FOR DUCT VELOCITY TO VELOCITY PRESSURE[15]

v_a, fpm	h_v, in. H_2O	v_a, fpm	h_v, in. H_2O
400	0.010	3,400	0.720
500	0.016	3,500	0.764
600	0.022	3,600	0.808
700	0.031	3,700	0.853
800	0.040	3,800	0.900
900	0.051	3,900	0.948
1,000	0.062	4,000	0.998
1,100	0.075	4,100	1.049
1,200	0.090	4,200	1.100
1,300	0.105	4,300	1.152
1,400	0.122	4,400	1.208
1,500	0.140	4,500	1.262
1,600	0.160	4,600	1.319
1,700	0.180	4,700	1.377
1,800	0.202	4,800	1.435
1,900	0.225	4,900	1.496
2,000	0.249	5,000	1.558
2,100	0.275	5,100	1.621
2,200	0.301	5,200	1.685
2,300	0.329	5,300	1.751
2,400	0.359	5,400	1.817
2,500	0.389	5,500	1.886
2,600	0.421	5,600	1.955
2,700	0.454	5,700	2.026
2,800	0.489	5,800	2.098
2,900	0.524	5,900	2.170
3,000	0.561	6,000	2.244
3,100	0.599	6,100	2.320
3,200	0.638	6,200	2.397
3,300	0.678		

where

f = friction loss in inches of water per 100 feet

V = velocity in fpm in duct

D = inside diameter of duct in inches

Actual values can be twice as high because duct internal surfaces are not ideally clean and smooth. Note that as duct diameter decreases and length of ductwork increases (as in a system with multiple miscellaneous pickup points), pressure drop can become significant. If the system fan is not designed to provide this pressure drop (for example, where miscellaneous pickup points have been added latter), the duct pressure drop will result in less suction at the hoods.

A rectangular duct can be converted to the circular equivalent in the following manner:

1. A = the duct cross-sectional area in square feet
2. P = the perimeter in feet
3. $R = \frac{A}{P}$ the hydraulic radius in feet
4. 12R = Conversion of R to inches, r
5. D = 4r = equivalent diameter in inches

- Elbows - Losses for 90-degree elbows are determined as equivalent resistance in feet of straight duct. For other elbow angles, use:

  60-degree elbow = 0.67 x loss for 90-degree elbow
  45-degree elbow = 0.5 x loss for 90-degree elbow
  30-degree elbow = 0.33 x loss for 90-degree elbow

  For radius of 1.5D:

$$\text{Equivalent feet} = 130 \left(\frac{D}{48}\right)^{1.175} \qquad \text{(Eq. 12)}$$

  For a radius of 2D:

$$\text{Equivalent feet} = 89 \left(\frac{D}{48}\right)^{1.171} \qquad \text{(Eq. 13)}$$

For radius of 2.5D:

$$\text{Equivalent feet} = 73 \left(\frac{D}{48}\right)^{1.17} \qquad \text{(Eq. 14)}$$

- Branch Entry - Losses due to branch entry are best expressed as the equivalent feet of straight duct of the same diameter. The equivalent length is added to the actual length of straight duct and the loss is computed from the earlier friction loss equation.

  For an entry angle of 30 degrees:

  $$Z = 20 \left(\frac{D}{20}\right)^{1.214} \qquad \text{(Eq. 15)}$$

  where

  Z = equivalent feet

  D = diameter in inches

  For an entry angle of 45 degrees:

  $$Z = 32 \left(\frac{D}{20}\right)^{1.248} \qquad \text{(Eq. 16)}$$

  All branches should enter the main duct at the large end of the transition, at an angle not to exceed 45 degrees, preferably 30 degrees or less. Branches should be connected only to the top or sides of the main duct, never to the bottom. Two branches should never enter a main at diametrically opposite points.

- Contraction and Expansion - When the cross-sectional area of a duct contracts, a pressure loss occurs. This loss is a function of the abruptness of the contraction. When the cross-sectional area expands, a portion of the decrease in velocity pressure becomes static pressure. The increases and decreases in pressure from expansion and contraction are calculated from equations in Reference 15:

The summation of the transport velocity requirements from all of these losses plus the hood or entry losses plus the control device and stack discharge loss determines the size and power of the fan. A pressure diagram can be useful in characterizing and understanding a given system. Figure 20 presents a hypothetical pressure diagram for a simple system.

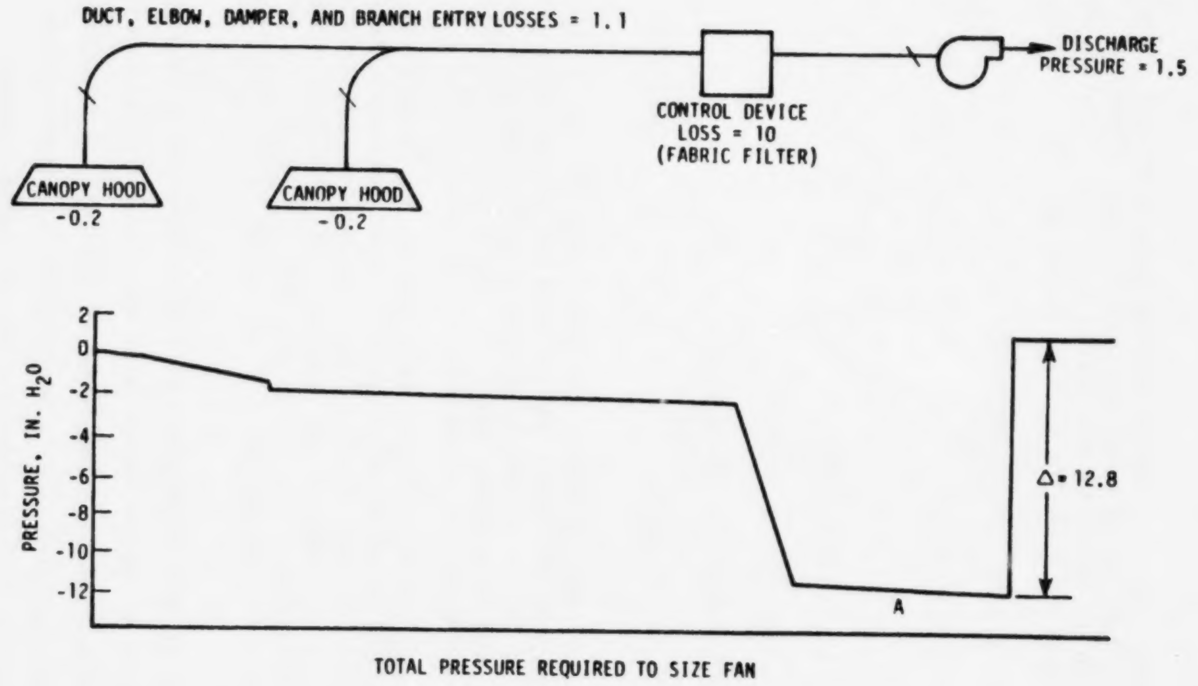

DUCT, ELBOW, DAMPER, AND BRANCH ENTRY LOSSES = 1.1
CANOPY HOOD
-0.2
CANOPY HOOD
- 0.2
CONTROL DEVICE
LOSS = 10
(FABRIC FILTER)
DISCHARGE
PRESSURE = 1.5
2
0
-2
-4
-6
-8
-10
-12
PRESSURE, IN. H_2O
A
Δ = 12.8
TOTAL PRESSURE REQUIRED TO SIZE FAN

Figure 20. Simple pressure diagram.

4.3 BRANCHED SYSTEMS

The addition of new or additional pickup points that are in turn tied into a main duct system has to be carefully designed to achieve the desired flow balance throughout the system. If duct sizes are not compatible with the static pressure throughout the system, the desired air flows will not be achieved. Thus, it is necessary to provide a means of distributing air flow between branches, either by balanced design or by the use of dampers. Figure 21 illustrates the widely used approach of tapered cross section for branch entry systems. This design maintains constant duct velocity in the main duct.

The two approaches for designing duct systems are 1) to balance duct diameters to match desired flows in each branch, and 2) to use dampers or blast gates to control flow. Many systems use a combination of both approaches.

Balanced design is theoretically preferable to minimize the tampering with dampers and the dependence on operating personnel. The design calculations begin at the branch of greatest resistance and proceed as follows: branch to main, section of main to section of main, main to control equipment on to the fan. At each junction point, the static pressure necessary to achieve the desired flow in one stream must match the static pressure in the other.

Damper adjustments permit the desired flow to be varied through each portion of the system after initial design. This permits adjustments to be made by trial and error to accommodate variation in actual conditions. The design calculations begin at the branch of greatest resistance, and pressure drops are calculated through the branches and duct sections to the fan. At each point where two gas streams meet, the combined flow is then used, and when it reaches the main duct, this combined flow is added to the main duct flow with no attempt to balance the static pressure in the joining gas streams. Branches are sized for the desired minimum duct or transport velocity at the desired flow rate.

Advantages and shortcomings of these two methods are further outlined in Table 5.

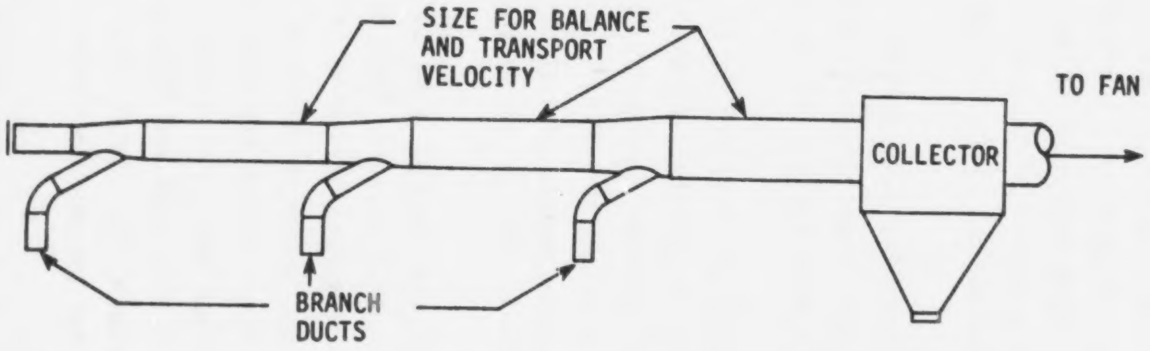

SIZE FOR BALANCE
AND TRANSPORT
VELOCITY
TO FAN
COLLECTOR
BRANCH
DUCTS

Figure 21. Taper duct system.

Courtesy: Industrial Ventilation.

TABLE 5. RELATIVE ADVANTAGES OF USING DUCT DAMPERS[14]

Balance without dampers	Balance with dampers
Air volumes cannot be easily changed by the operator.	Air volumes may be changed relatively easily.
Small degree of flexibility for future equipment changes or additions; the ductwork is "tailor-made" for the job.	Greater degree of flexibility for future changes or additions.
Choice of exhaust volumes for a new unknown operation may be incorrect; in such cases some ductwork revision is necessary.	Correction of improperly estimated exhaust volumes is easy within certain ranges.
No unusual erosion or accumulation problems.	Partially closed dampers or blast gates may cause erosion and thereby change the degree of restriction or cause accumulations of material.
Ductwork will not plug if velocities are chosen wisely.	Ductwork may plug if persons have tampered with the blast gate adjustment.
Total air volumes are slightly greater than design air volumes because of the additional air handled to achieve balance.	Balance may be achieved with design air volume.
Poor choice of "branch of greatest resistance" will show up in design calculations.	Poor choice of "branch of greatest resistance" may remain undiscovered. In such case the branch or branches of greater resistance will be "starved."
Layout of system must be in complete detail, with all obstructions cleared and length of runs accurately determined. Installations must follow layout exactly.	Leeway is allowed for moderate variation in duct location to miss obstructions or interferences not known at time of layout.

SECTION 5

FAN SYSTEMS

Centrifugal fans are normally used in conjunction with large emission control systems because of the large flow rates and high pressure drops in these systems. Although some axial fans can deliver large volumes of air at high resistance, they are best suited for clean-air applications. The presence of dust causes rapid erosion of axial fan components because of the high tip speed of the fan and the high air velocity through the fan housing. Centrifugal fans can be designed for differing gas characteristics encountered in various emission control applications. This section discusses the major aspects of centrifugal fans with regard to their applicability for large ventilation systems.

5.1 FAN TYPES AND OPERATING CHARACTERISTICS

A centrifugal fan is used to transfer energy to gases by centrifugal action. Figure 22 shows the components of the centrifugal fan and layout of a typical industrial fan system. It consists of a wheel or rotor that is rotated by an electric motor in a scroll-shaped housing. The gases enter the housing axially, make a right-angle turn, and are forced through the blades of the rotor and into the housing by centrifugal force. The centrifugal force imparts velocity pressure to the air, and the diverging shape of the scroll converts a portion of the velocity head into static head.

Centrifugal fans are classified according to the following blade configurations:

- Backward-curved blade
- Forward-curved blade
- Straight blade

Figure 23 shows a few basic centrifugal fan blade configurations and impeller arrangements.[16]

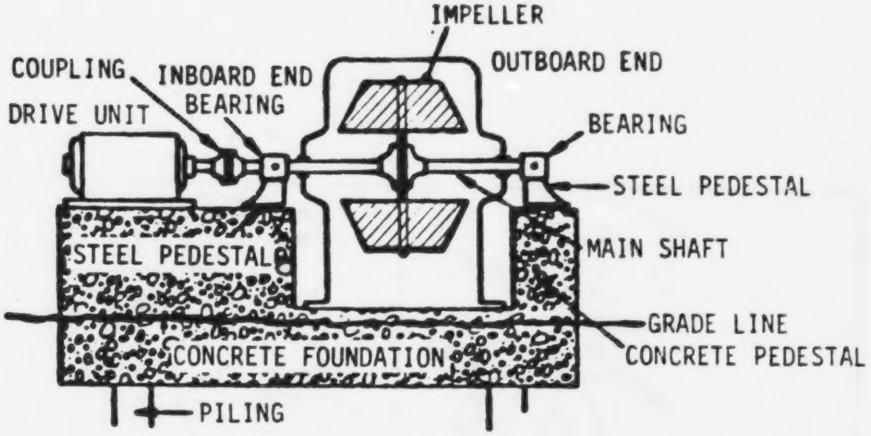

IMPELLER
COUPLING
INBOARD END
BEARING
OUTBOARD END
DRIVE UNIT
BEARING
STEEL PEDESTAL
STEEL PEDESTAL
MAIN SHAFT
GRADE LINE
CONCRETE FOUNDATION
CONCRETE PEDESTAL
PILING

Figure 22a.

Courtesy: Hydrocarbon Processing, June 1975. Article by J. W. Martz and R. R. Pfahler entitled "How to Troubleshoot Large Industrial Fans."

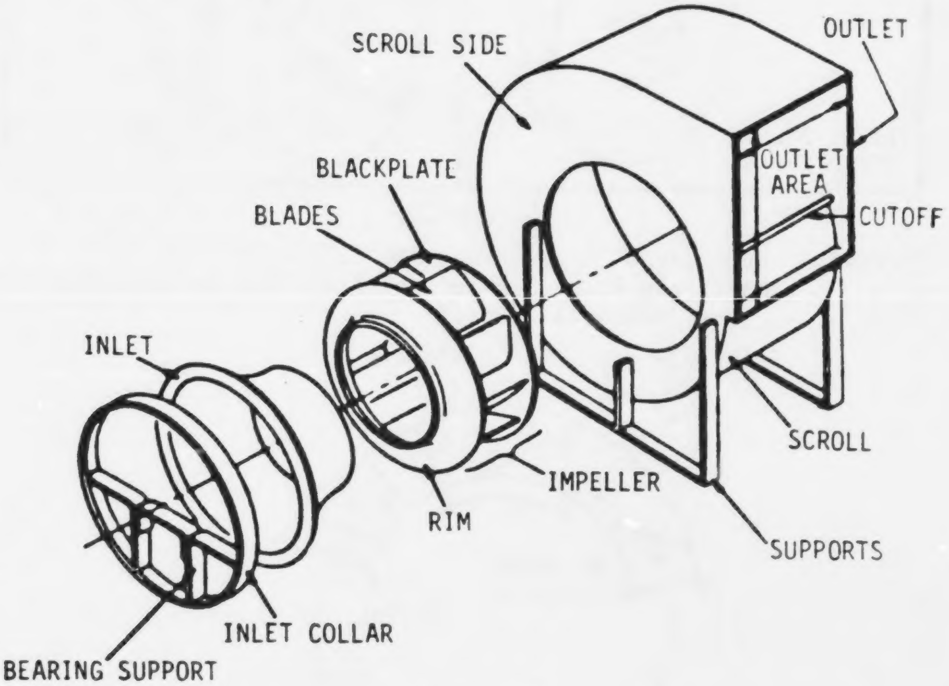

SCROLL SIDE
OUTLET
OUTLET AREA
BLACKPLATE
CUTOFF
BLADES
INLET
SCROLL
IMPELLER
RIM
SUPPORTS
INLET COLLAR
BEARING SUPPORT

Figure 22b.

Courtesy: "With permission of the American Society of Heating, Refrigerating & Air Conditioning Engineers, Inc., Atlanta, GA.

Figure 22. Centrifugal fan components and layout of a typical industrial fan system.

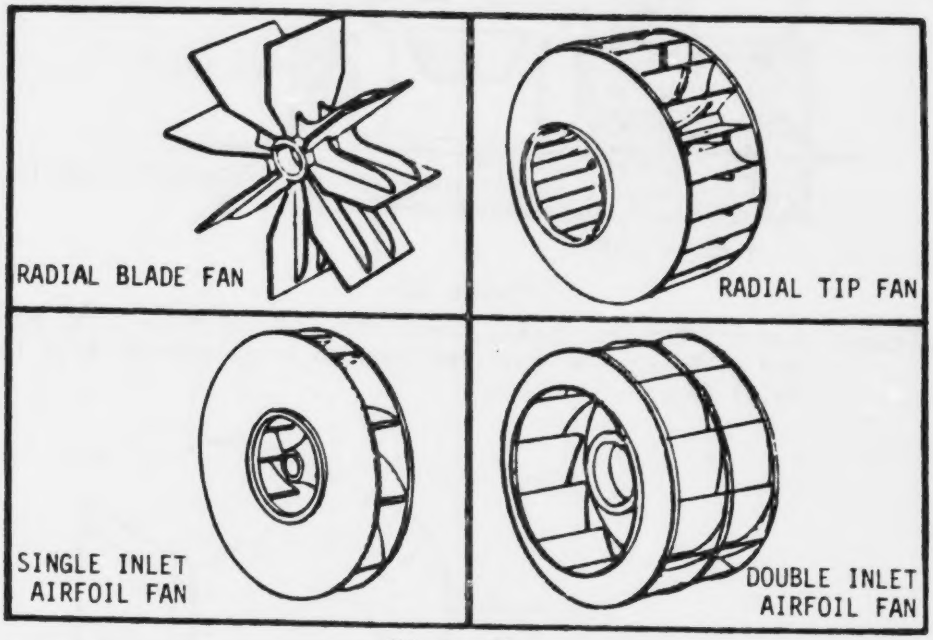

RADIAL BLADE FAN
RADIAL TIP FAN
SINGLE INLET AIRFOIL FAN
DOUBLE INLET AIRFOIL FAN

Figure 23a.

Courtesy: Hydrocarbon Processing, June 1975. Article by J. W. Martz and R. R. Rfahler entitled "How to Troubleshoot Large Industrial Fans."

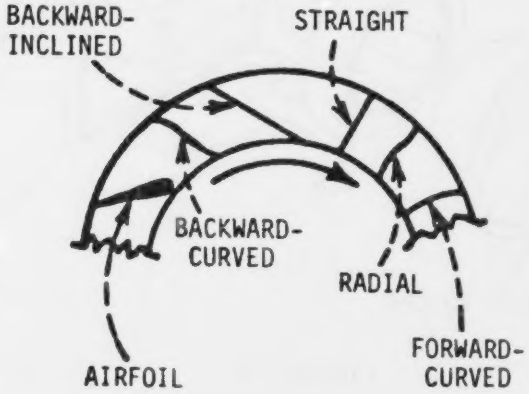

BACKWARD-INCLINED
STRAIGHT
BACKWARD-CURVED
RADIAL
FORWARD-CURVED
AIRFOIL

Figure 23b.

Courtesy: Excerpted by special permission from CHEMICAL ENGINEERING (date of issue) Copyright (c) (year), by McGraw-Hill, Inc., New York, N.Y. 10020.

Figure 23. Centrifugal fan blade configurations and impeller arrangements.

The size, shape, and number of blades affect the operating characteristics of the fan. Fan performance is characterized by the volume of gas flow, pressure, fan speed, power requirement, and operating efficiency. The relationship of these parameters is measured according to the testing methods sponsored by the National Association of Fan Manufacturers or the American Society of Mechanical Engineers. The fan is tested from shutoff conditions to free-delivery conditions. At shutoff, the duct is completely blanked off; at free delivery, the outlet resistance is zero. Between these two conditions, various flow restrictions are placed at the end of the duct to simulate various operating conditions. The operating parameters are measured at each test point and plotted against volume on the abscissa. Figure 24 illustrates the fan testing procedure and shows the typical fan characteristic curves.

Each fan type has a different performance characteristic. The fan performance curves are used in the selection of a fan type. Generally, the characteristics of geometrically similar fans are identical. The fan manufacturers can predict the performance of a large fan from the tests on a smaller but geometrically similar fan.

5.1.1 Backward-Curved-Blade Fan

The blades in a backward-curved-blade centrifugal fan are inclined in a direction opposite to the direction of rotation. The blades (usually 14 to 24) are supported by a solid steel backplate and shroud ring. The scroll-type housing permits efficient conversion of velocity head into static head.

The characteristics of a backward-curved-blade fan are shown in Figure 25. The static pressure of this fan rises sharply from free delivery to about 50 percent volume point. Beyond this point and up to the no-delivery point the pressure remains approximately constant. Maximum efficiency occurs at maximum horsepower input. The horsepower requirement is self-limiting; it rises to a maximum as the capacity increases and then decreases with additional capacity. This self-limiting horsepower characteristic of the backward-curved-blade centrifugal fan prevents overloading of the motor when the fan load exceeds its design capacity. The operating efficiency of the backward-curved-blade fan is high, and this fan develops higher pressure than the forward-curved-blade fan.

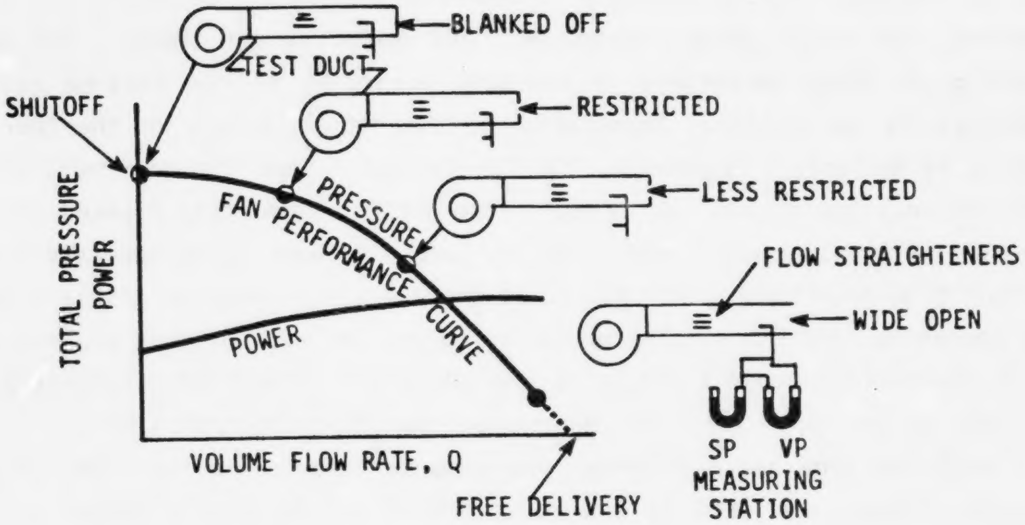

BLANKED OFF
TEST DUCT
SHUTOFF
RESTRICTED
LESS RESTRICTED
FLOW STRAIGHTENERS
WIDE OPEN
TOTAL PRESSURE
POWER
PRESSURE
FAN PERFORMANCE CURVE
POWER
VOLUME FLOW RATE, Q
FREE DELIVERY
SP
VP
MEASURING STATION

Figure 24a.

Courtesy: ASHRAE Handbook. Equipment Volume.

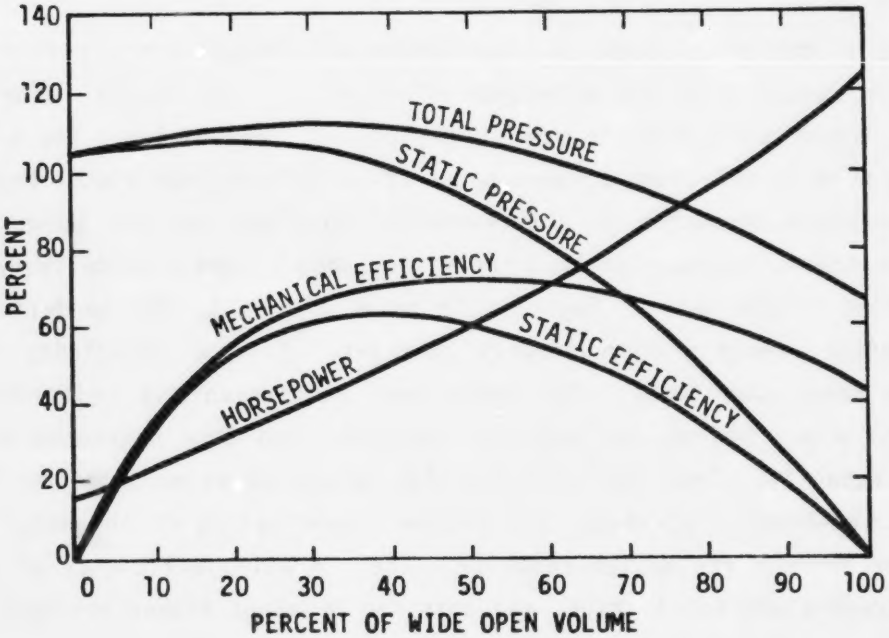


Figure 24b.

Source: U.S. Environmental Protection Agency. Standards Support Documents: An Investigation of the Best Systems of Emission Reduction for Electric Arc Furnaces in the Steel Industry. (Draft) June 1974.

Figure 24. Fan testing procedure and typical characteristic curves.

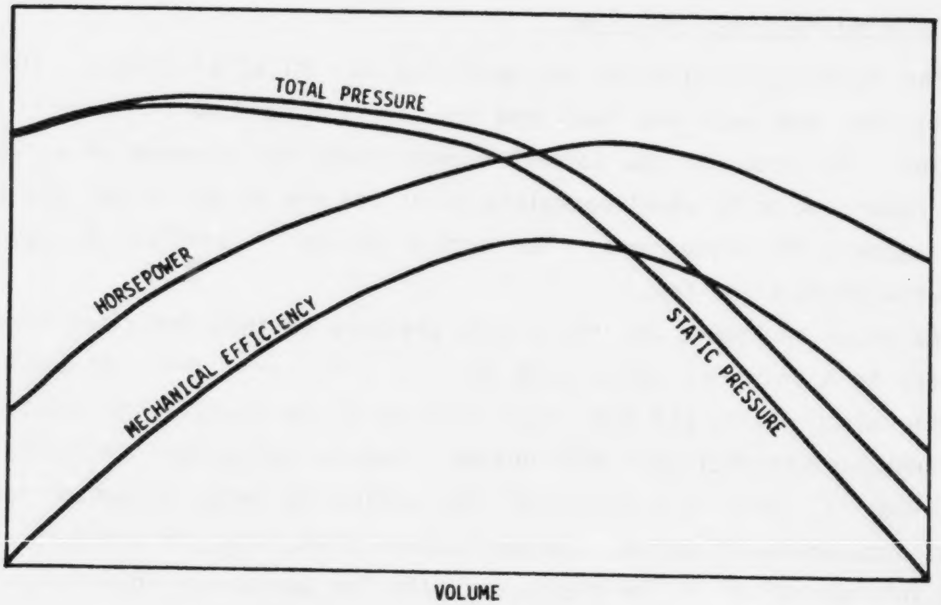


Figure 25. Typical characteristic curves for a backward-curved-blade centrifugal fan.

Because the backward-curved blades are conducive to buildup of material, they are not recommended for dirty streams. These fans are generally used in ventilating applications where large volumes of clean air are to be handled on a continuous basis. When used for emission control applications, the backward-curved-blade fan must be installed on the clean-air side of the control system.

5.1.2 Forward-Curved-Blade Fan

The forward-curved-blade fan generally has 20 to 64 blades. The blades are shallow, and both the heel and tip are curved toward the direction of rotation. The rotor of the forward-curved-blade fan is known as a "squirrel-cage" rotor. A solid steel backplate holds one end of the blade, and a shroud ring supports the other end. The scroll design is similar to that of the backward-curved-blade fan.

As shown in Figure 26, the static pressure of this fan rises from a free delivery to a point at approximate maximum efficiency, drops to about the 25 percent volume point, and then rises back up to the no-delivery point. Horsepower requirement increases with volume. Because horsepower increases rapidly with capacity, there is a danger of overloading the motor if system resistance is not accurately estimated. Forward-curved-blade fans are designed to handle large volumes of air at low pressures. The fan speeds are relatively low, and the pressures developed by forward-curved-blade fans are generally insufficient for emission control system applications. These fans are used extensively in heating, ventilating, and air conditioning applications.

5.1.3 Straight-Blade Fan

Straight-blade fans are the simplest of all centrifugal fans. The fan usually has 5 to 12 blades, which are generally attached to the rotor by a solid steel backplate or a spider built up from the hub. The rotors are relatively large in diameter.

Figure 27 shows the performance characteristics of the straight-blade fan. The static pressure of this fan rises sharply from free delivery to a maximum point near no delivery, where it falls off. Mechanical efficiency rises rapidly from no delivery to a maximum near maximum pressure, and then drops slowly as the fan capacity approaches free delivery.

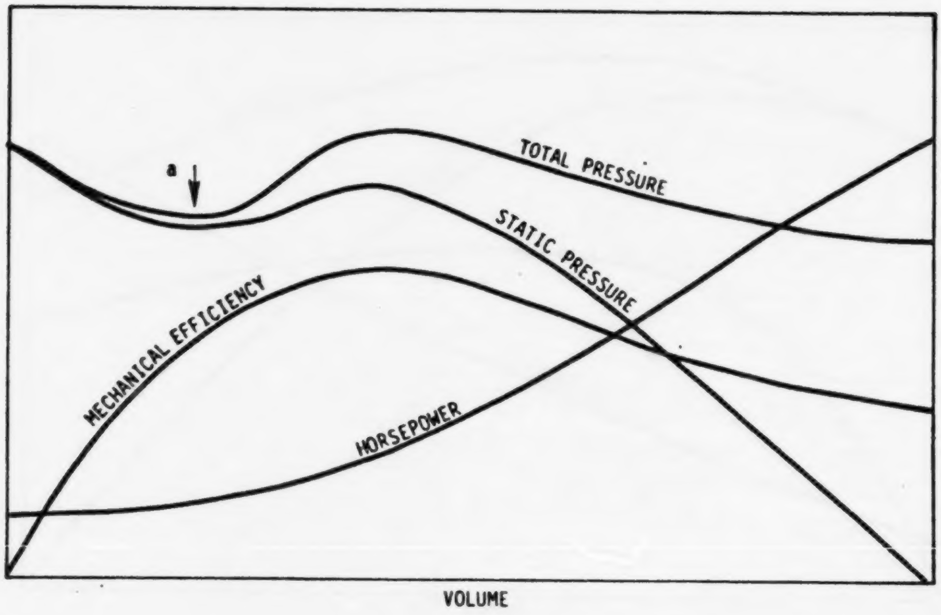


Figure 26. Typical characteristics curves for a forward-curved-blade fan.

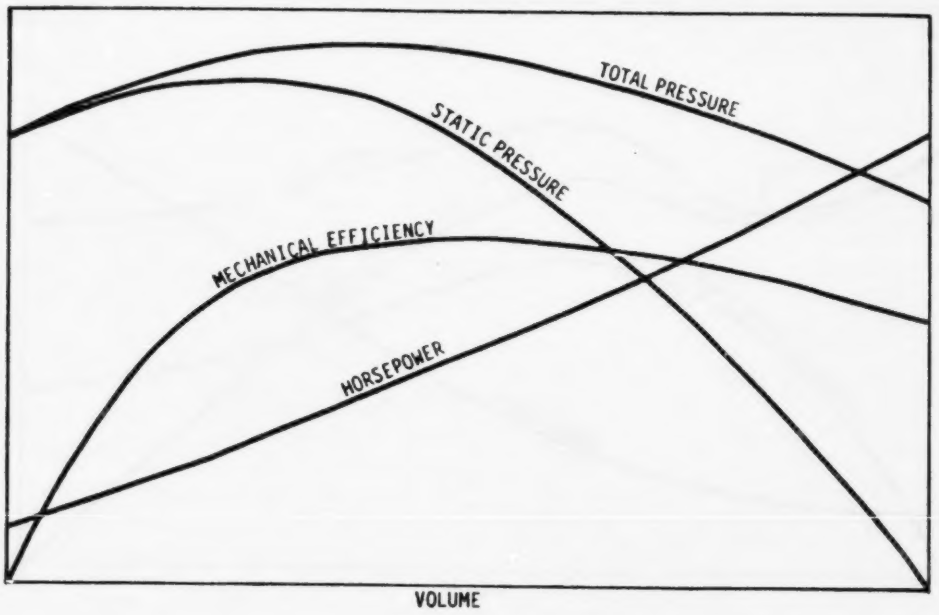


Figure 27. Typical characteristics curves for a straight-blade fan.

The straight-blade fan can be used in exhaust systems handling gas streams that are contaminated with dusts and fumes. Various blade designs and scroll designs have been developed for specific dust-handling applications.

5.1.4 Backward-Inclined Blade Fan

The two types of backward-inclined blades in the centrifugal fan class are air foil blades and flat blades.

5.1.5 Rim-Type Wheel Fans

With abrasive material, or material that tends to stick, a rim-type wheel provides more structural integrity; some of these have back plates on them. Literally dozens of rim-type fans are on the market. Each fan typically has at least six blades. The radial tip tends to develop more static pressure so it can develop the same gas flow as the straight-blade radial at higher pressure drop.

5.2 FORCED VERSUS INDUCED DRAFT

The terms "forced draft" and "induced draft" come from boiler technology, where they refer to either forcing air through the boiler (with a blower) or pulling air through the boiler with a fan located on the exhaust side. Figure 28 illustrates the meaning of these terms in the context of air pollution control ventilation systems. Essentially, the control device takes the place of the boiler. Two considerations are important in the use of forced-draft systems:

1. Whether the fan is exposed to cleaned gas or dirty gas
2. Whether the control device is under pressure or suction

The suitability of the centrifugal fan for dirty applications has already been discussed. The axial fan would clearly only be suitable in an induced-draft system. The forced-draft system is generally preferred in large applications, particularly those controlled by a fabric filter, because control costs are lower, even when the higher fan cost and maintenance are taken into account. Many large (i.e., 300,000 cfm and greater) systems treat relatively clean gas because the process gas has been diluted with indraft air. Thus,

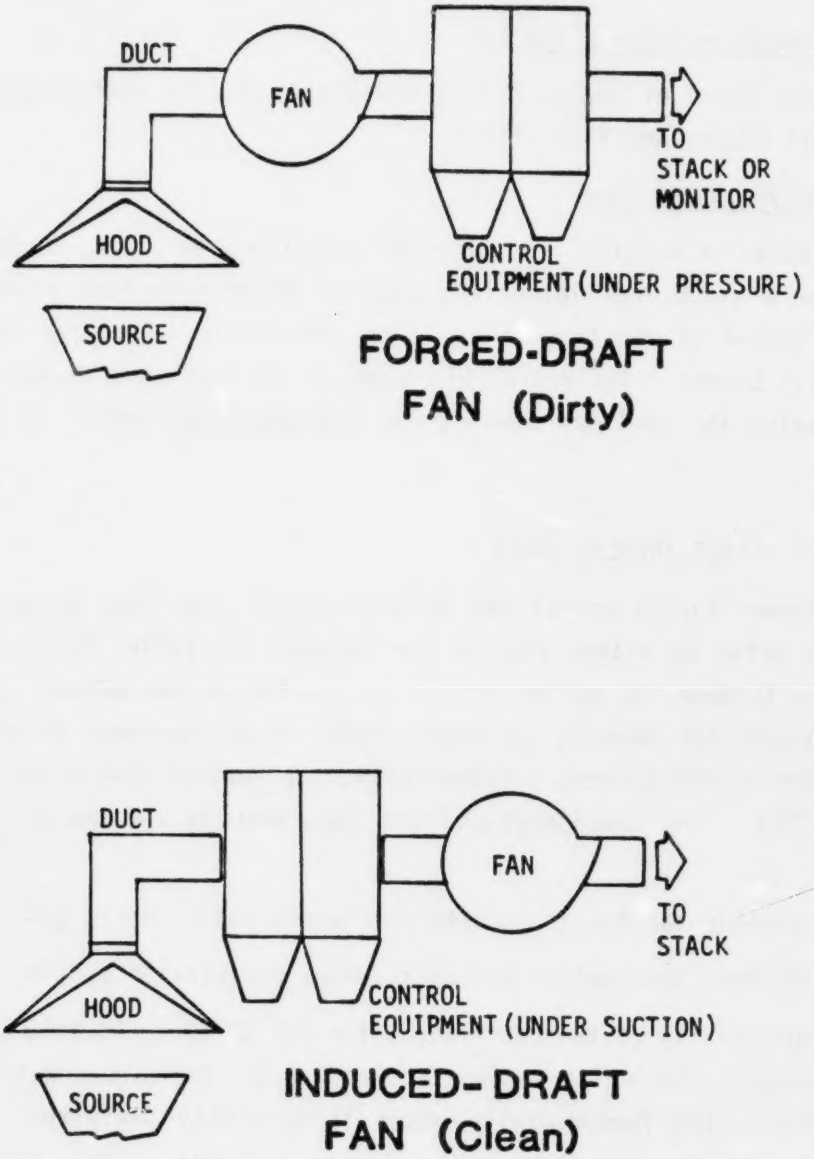

DUCT
FAN
TO
STACK OR
MONITOR
HOOD
CONTROL
EQUIPMENT (UNDER PRESSURE)
SOURCE
FORCED-DRAFT
FAN (Dirty)
DUCT
FAN
TO
STACK
HOOD
CONTROL
EQUIPMENT (UNDER SUCTION)
SOURCE
INDUCED-DRAFT
FAN (Clean)

Figure 28. Basic principle of induced versus forced draft.

even the "dirty" gas inlet to the fan is relatively low in dust concentrations, i.e., 1.0 gr/scf or less. Also, this type of system does not usually require a stack. Systems that use scrubbers or electrostatic precipitators, on the other hand, usually have induced-draft fans. In scrubber systems with variable pressure drops, the induced-draft fan is especially needed to control pressure drop.

Forced-draft fans that are used on the "dirty" gas side are usually larger and rotate at a slower speed than fans on the clean side to cut down on the abrasion of the fan blades.

5.3 FAN REQUIREMENTS FOR EMISSION CONTROL SYSTEM APPLICATIONS

Basically, a fan can develop static pressure without delivering much gas volume, or it can deliver very little static pressure at high gas volume. Unfortunately, it cannot do both at the same time. Any given fan cannot perform beyond the limitation of its operating curve.

Increasing the rotation speed and the gas volume through the system doesn't shift the system resistance curve, but it permits the fan to overcome more resistance and move more gas through the system by shifting the fan operating curve. A fan is basically a volumetric energy machine. A certain gas volume is carried between each blade. Therefore, if the number of times the blades pass the outlet is increased, more gas is moved and at an increase in pressure. Any increase in fan rotation speed increases fan horsepower requirements. Two ways to control capacity are 1) to put in dampers to waste static pressure across the damper to reduce gas flow, and 2) to reduce fan rotation speed. If dampers are added, the fan is still expending energy to develop pressure drop; therefore, reducing fan rotation speed is more efficient because it develops a lower horsepower and uses less energy than a damper. Table 6 summarizes the basic fan laws.

In the design of fans for emission control system applications, considerations must be given to the nature of the gases being handled. Emission control system fans may be subjected to one or more of the following operating conditions:

TABLE 6. BASIC FAN LAWS

Variable	When speed changes	When density changes
Volume	Varies directly with speed ratio $CFM_2 = CFM_1 \left(\frac{RPM_2}{RPM_1}\right)$	Does not change
Pressure	Varies with square of speed ratio $P_2 = P_1 \left(\frac{RPM_2}{RPM_1}\right)^2$	Varies directly with density ratio $P_2 = P_1 \left(\frac{D_2}{D_1}\right)$
Horsepower	Varies with cube of speed ratio $HP_2 = HP_1 \left(\frac{RPM_2}{RPM_1}\right)^3$	Varies directly with density ratio $HP_2 = HP_1 \left(\frac{D_2}{D_1}\right)$

- High temperatures
- Corrosive gases
- Dust-laden gases
- Presence of abrasive particles
- Presence of explosive materials in the gases

Special construction materials can provide protection against high temperatures and corrosive properties. Bronze alloys are used for handling sulfuric acid fumes and other sulfates, halogen acids, various organic gases, and mercury compounds. Stainless steel is the most commonly used corrosion-resistant metal for impellers and fan housings. Protective coatings such as bisonite, cadmium plating, hot galvanizing, and rubber covering provide resistance to corrosive gases. Depending on the particular application, soft, medium, or firm rubber can be bonded to the metal. A good bond will yield an adhesive strength of 700 pounds per square inch. Rubber-covered fans have proved exceptionally durable.

When a fan must handle explosive gases, the construction material should be such that it does not produce a spark if accidentally struck by other metal--e.g., bronze and aluminum alloys. In most applications, it is preferable to combust explosive gases prior to entering the ductwork. In the case of a BOF controlled by the closed-hood system, however, the gas is not combusted, so the inherent heating value rising from the contained carbon monoxide can be retained.

Fans handling dust-laden gases must be protected from wear due to abrasion. When wear is expected, some manufacturers equip their fans with wear strips and weld beads. The strips consist of a thick, cross-hatched hardened floor plate welded to the blade at the centerplate or backplate, and the plate is built up at the edges with weld beads. The inertia of the dust particles carries them toward the backplate or centerplate, where the wear plate withstands most of the abrasion. The weld beads, which are angled along the edge of the wear plate, break up the particle flow and prevent impingement and scouring. For severe abrasion conditions, fans can be equipped with full-blade liners or heavy-duty rotors with thick wear plates bolted or welded to the full face of the blades. The fan construction is such that worn wear plates can be replaced on site.

5.4 FAN ARRANGEMENTS

Whenever possible, the fan should be installed on the clean side of the emission control system, where it will be subjected to less severe operating conditions. Plants with multiple emission control systems generally have separate fans for each control system. If a fan is shared by more than one emission control system, fan operation can be difficult to control. At such installations, the fan may operate at less than optimum conditions because it may be subjected to fluctuating loads.

For applications where fan load is expected to vary widely and the peak fan load is relatively large, multiple fans may provide better control over the system operation. A parallel fan system will increase reliability and flexibility. The performance of two fans in parallel can be predicted by combining the ordinates and abscissas of the pressure-volume curves of both fans. At low loads, the parallel system can also be operated as a single fan system. Individual fan loads can be adjusted to optimize the performance of the system at a given load.

5.5 FAN DRIVES

Large centrifugal fans are generally driven by three-phase, alternating-current motors. The two principal types of motors used for driving fans are 1) the squirrel-cage induction motor, and 2) the wound-rotor induction motor. Although both motor types are self-starting, special starting controls are needed in many applications to limit starting time to acceptable values. Both types operate at rated load with very little slip.

The squirrel-cage motor takes its name from the rotor construction. Among the various standard designs, the one designated as Design B is usually used on fans. This motor is suitable for continuous operation at rated load, and its starting current is relatively low.

Wound-rotor motors, also known as slip-ring motors, can provide adjustable-speed drive if desired. Speed reductions below 50 percent are not recommended, however. Top speed and efficiency are about the same as for the squirrel-cage motor.

5.6 FAN CONTROLS

The throughput of fans may be changed by varying the speed or by changing the pressure condition at the inlet and/or outlet. Variable inlet vanes provide control on the inlet side of the fan, and dampers can be used on either the inlet or outlet side. Each of the fan control methods entails a loss in efficiency over most of the operating range. Because most large-capacity fans are ordinarily driven by constant-speed motors, it is usually not possible to control the fan by varying the speed. Variable inlet vanes can provide gradual load adjustment; however, these must be purchased with the fan. Dampers offer flexibility of location and control. A fan system with dampers can be designed to meet a wide range of load fluctuations. Table 7 lists the basic damper types, and the following subsections provide brief descriptions of the major types.

5.6.1 Louvre

Louvre or multiblade dampers may be of the opposed-blade or the parallel-blade design (Figures 29a and 29b). An overall view of a parallel-blade, multilouvre damper is shown in Figure 29c. Parallel-blade louvre dampers can be closed tight, but they offer little modulation ability. Conversely, opposed-blade louvre dampers offer excellent modulation ability, but they cannot be closed as tightly as the parallel-blade louvre dampers can. Louvre dampers may be used to regulate and isolate gas flow. For isolation, two dampers can be used together and sealed by pressurizing the chamber formed by the ductwork between the dampers with a seal-air fan. A single damper may be used for gas flow regulation.

5.6.2 Guillotine Damper

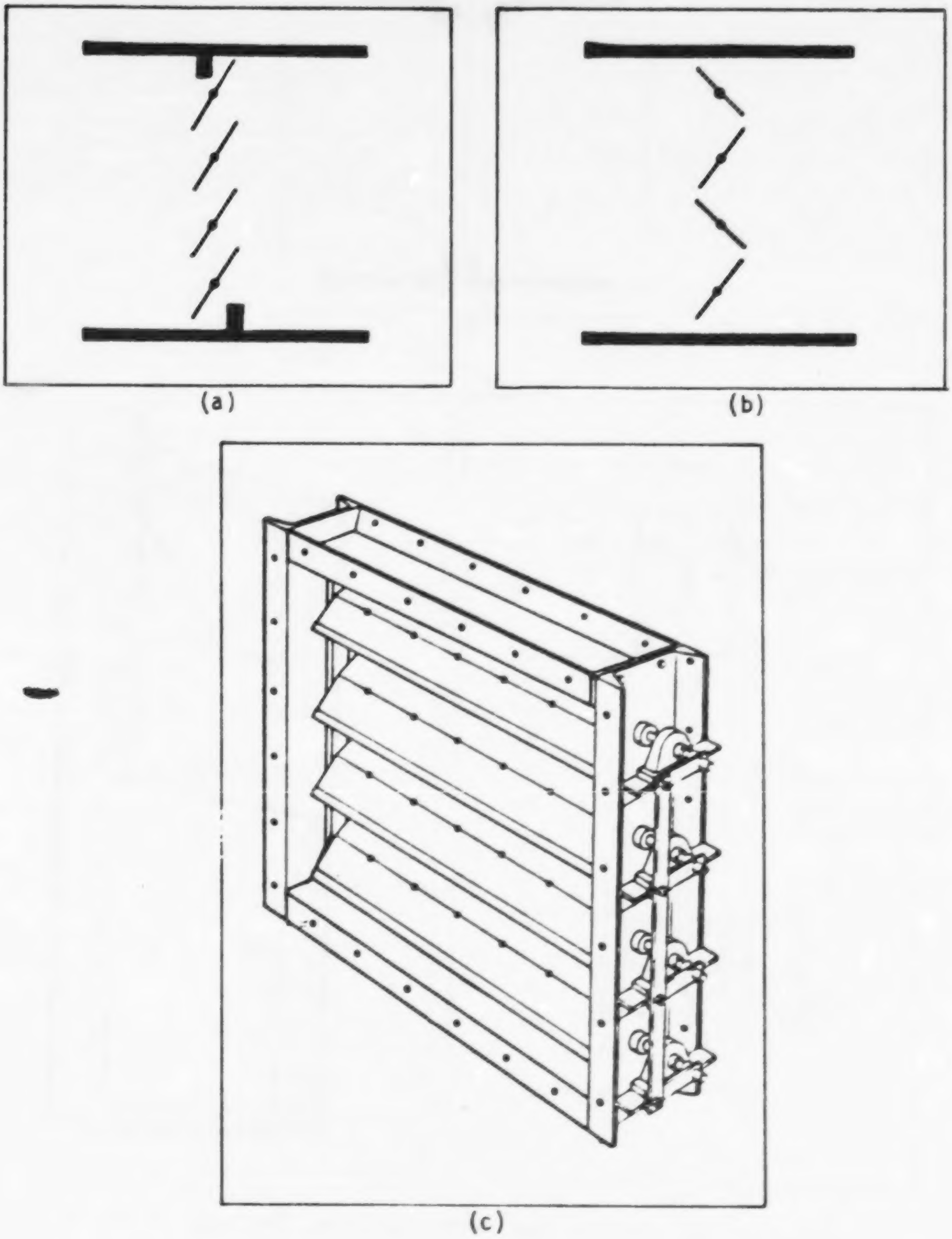
A guillotine damper may be a top-entry or bottom-entry design, with or without seal air (Figure 30). Guillotine dampers for system isolation may be equipped with a seal-air blower, single- or double-bladed, to pressurize the sealing space and thus ensure against gas leakage past the damper.

5.6.3 Butterfly Damper

Butterfly dampers are often used for secondary duct runs (Figure 31). They are mounted by a center shaft that crosses the duct, and the damper plate

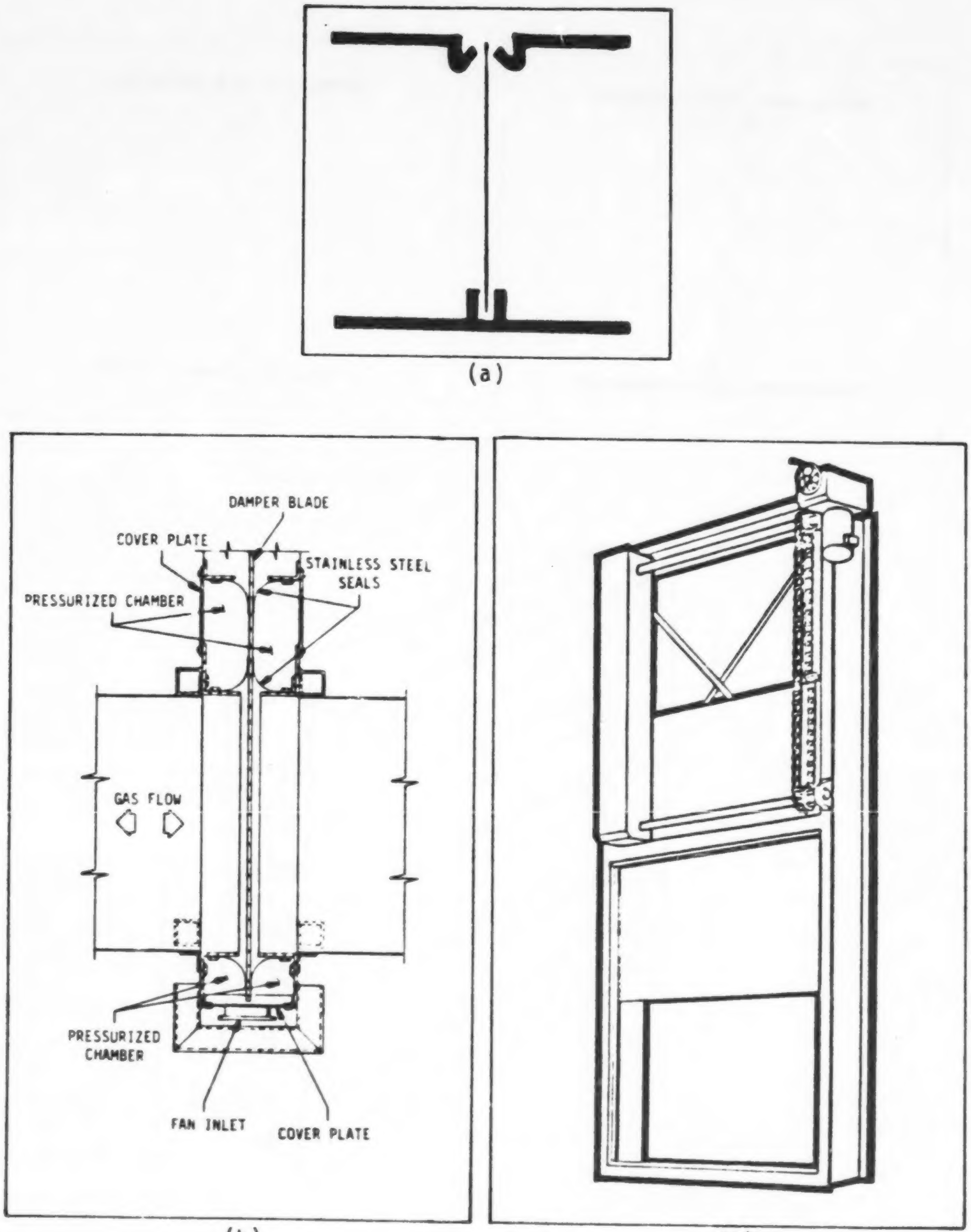
TABLE 7. BASIC DAMPER TYPES

Generic	Specific	Common designs
Louvre	Parallel-blade multilouvre	Single louvre Double louvre Double louvre/seal-air
	Opposed-blade multilouvre	Single louvre Double louvre Double louvre/seal-air
Guillotine	Top-entry guillotine Top-entry guillotine/seal-air Bottom-entry guillotine Bottom-entry guillotine/seal-air	
Butterfly		
Blanking plate		



Courtesy: Frisch Division, DAYCO Company, Chicago, Illinois.

Figure 29. Louvre damper: (a) parallel-blade multilouvre; (b) opposed-blade multilouvre; (c) view of parallel-blade, multilouvre damper showing linkage.


DAMPER BLADE
COVER PLATE
STAINLESS STEEL SEALS
PRESSURIZED CHAMBER
GAS FLOW
PRESSURIZED CHAMBER
FAN INLET
COVER PLATE

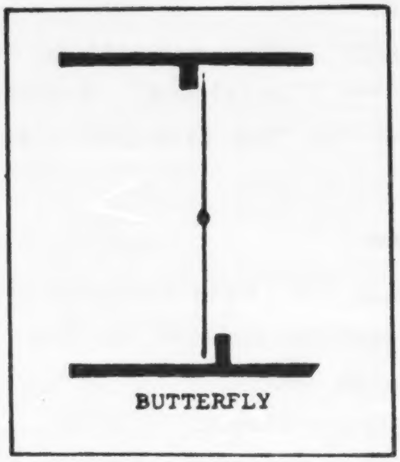
(a)

(b)

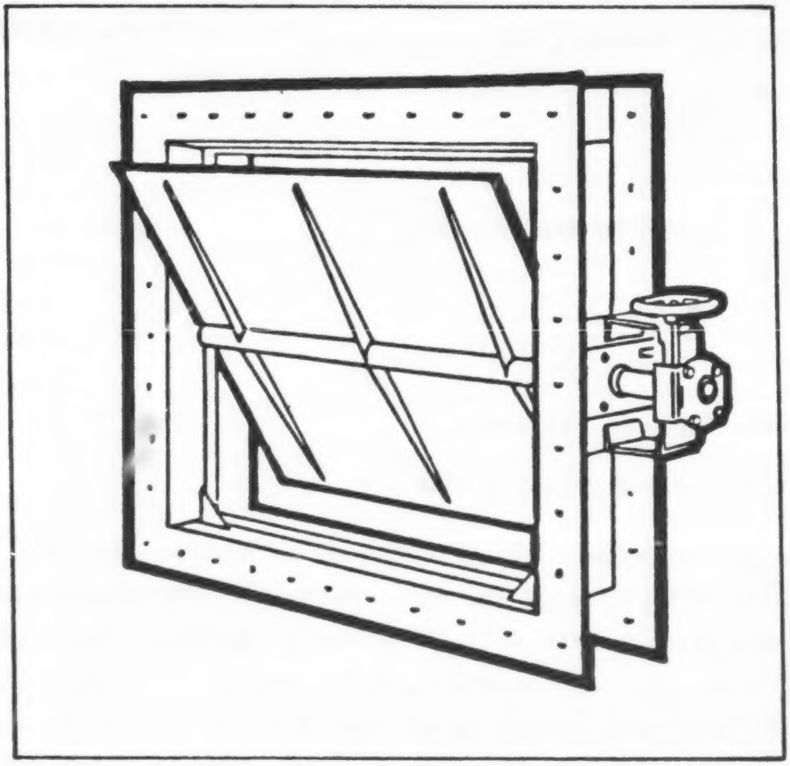
(c)

Courtesy: Frisch Division, DAYCO Company, Chicago, Illinois.

Figure 30. Guillotine damper: (a) simplified cross-sectional view of a guillotine damper; (b) guillotine isolation damper using seal air; (c) top-entry type guillotine damper, showing operation.



(a)



(b)

Courtesy: Frisch Division, DAYCO Company, Chicago, Illinois.

Figure 31. Butterfly damper: (a) simplified cross-sectional view of a butterfly damper; (b) butterfly damper showing hand operator.

rotates about the shaft from a plane parallel to the gas flow (open) to a plane perpendicular to the gas flow (closed). Butterfly dampers are used most often to regulate gas flow, but they have been used for opening and closing off ducts and fan systems.

5.6.4 Blanking-Plate Damper

The most basic damper is the simple blank-off plate. Blanking plates are essential when a process must be isolated for the protection of the maintenance crew. Should it become necessary for persons to enter any section of the ductwork, the blanking plate ensures isolation of that section. When used in conjunction with positive-ventilation air purge, the blanking plates ensure the safety of personnel.

Blanking plates are similar to guillotine dampers in that they cut across the duct opening; however, the track design for a blanking plate is intended only to guide the plate as it is put in place and bolted down.

5.7 FAN SIZING

The air (base) horsepower requirement of a fan can be calculated by Equation 17:

$$\text{Air hp} = Qh/6356 \qquad \text{(Eq. 17)}$$

where Q = inlet volume, ft^3/min

h = total static pressure rise, in. H_2O

Fan brake horsepower can be obtained by dividing the air horsepower by the mechanical efficiency of the fan. Mechanical efficiency for most centrifugal fan operating points will be 50 to 65 percent. Fan size can also be determined from the fan manufacturer's catalog. Manufacturers' catalogs generally include multirating tables that give the operating parameter ranges for different fan models. Many larger fans, however, are custom-built and thus not found in multirating tables. If the inspector needs to do a detailed analysis on a custom-built fan, he/she must obtain the fan ratings and performance curves from either the manufacturer or the plant.

The multirating tables are generally based on standard conditions. When air is not at standard conditions, corrections must be made to volume, pressure, and horsepower corrections to select a fan at an "equivalent" volume and pressure. For a manufacturer's rating table to be used, the fan requirements must be converted to the density used in the ratings. The density correction is generally made by using the ratio of air density at standard conditions to the actual density of the air at the fan inlet. Table 8 provides air density correction factors at various temperatures and altitudes. Table 9 lists the densities of various common gases. It is usually sufficient to assume the density of air because it is the predominant gas, but in some cases (e.g., saturated gas from a scrubber), a density correction for composition may be necessary. A further density correction is theoretically required because the air at the fan inlet is under suction, which lowers the density. For example, the correction factor from -60 in. H_2O to sea level is about 1.18. Before a fan is chosen from the multirating tables, the following adjustments should be made:

1. Determine density factor (d = 0.075 for air at 70°F and 29.92 in. Hg)

2. If the gas flow is indicated at standard conditions, convert it to actual fan conditions:

$$Q = \left(\frac{14.7}{A}\right) \times \left(\frac{460 + T}{530}\right) \times \text{Vscfm}$$

where Q = actual volume of air entering the fan
A = barometric pressure corresponding to fan site altitude, psia
T = inlet temperature, °F
Vscfm = volume of air at standard conditions (70°F and 14.7 psia)

3. Multiply static pressure by the density factor, d

4. Using corrected static pressure and actual gas flow, Q, select fan from multirating tables

5. Divide the fan bhp selected in Step 4 by the density factor.

TABLE 8. AIR DENSITY CORRECTION FACTOR, d

Altitude, ft.		-1000	Sea level	1000	2000	3000	4000	5000	6000	7000	8000	9000	10,000
Barometer, in.	Hg Wg	31.02 422.2	29.92 407.5	28.86 392.8	27.82 378.6	26.82 365.0	25.84 351.7	24.90 338.9	23.98 326.4	23.09 314.3	22.22 302.1	21.39 291.1	20.58 280.1
Air temp., °F	-40	1.31	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.93	0.90	0.87
	0	1.19	1.15	1.11	1.07	1.03	0.99	0.95	0.91	0.89	0.85	0.82	0.79
	40	1.10	1.06	1.02	0.99	0.95	0.92	0.88	0.85	0.82	0.79	0.76	0.73
	70	1.04	1.00	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.74	0.71	0.69
	100	.98	0.95	0.92	0.88	0.85	0.81	0.78	0.75	0.73	0.70	0.68	0.65
	150	.90	0.87	0.84	0.81	0.78	0.75	0.72	0.69	0.67	0.65	0.62	0.60
	200	.83	0.80	0.77	0.74	0.71	0.69	0.66	0.64	0.62	0.60	0.57	0.55
	250	.77	0.75	0.72	0.70	0.67	0.64	0.62	0.60	0.58	0.56	0.58	0.51
	300	.72	0.70	0.67	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.48
	350	.68	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.51	0.49	0.57	0.45
	400	.64	0.62	0.60	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.42
	450	.60	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.40
	500	.57	0.55	0.53	0.51	0.49	0.47	0.45	0.44	0.43	0.41	0.39	0.38
	550	.54	0.53	0.51	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.36
	600	.52	0.50	0.48	0.46	0.45	0.43	0.41	0.40	0.39	0.37	0.35	0.34
	700	.47	0.46	0.44	0.43	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.32
	800	.44	0.42	0.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29
	900	.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27
	1000	.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27	0.26	0.25

Standard air density, sea level, 70°F = 0.075 lb/ft^3.

TABLE 9. DENSITY OF COMMON GASES

Gas	lb/ft^3
Hydrogen	0.0052
Oxygen	0.0828
CO	0.072
CO_2	0.1146
N_2	0.0728
Benzene	0.2017
Ammonia	0.0446
SO_2	0.1697
Water vapor	0.0466
Air	0.075

The fan brake horespower obtained from the above calculations is divided by the motor efficiency to obtain motor horsepower. Motor efficiency for three-phase motors such as those used on large systems is typically 85 to 90 percent.

Usually, fan horsepower is designed for so-called cold-start conditions, in which case the density correction factor is not used. This enables the motor and fan to pull the full quantity of "cold" (i.e., ambient) dry air upon startup without overload. After startup, when the air reaches operating conditions of temperature and humidity, it is less dense and the horsepower load is reduced.

SECTION 6

VENTILATION SYSTEM INSPECTION

Careful preparation and planning are vital to a successful inspection and evaluation of ventilation systems. An inspection will be meaningful only if the inspector knows what information he/she wants to collect and is familiar with the equipment at the site. Time invested in a file review will reduce the inspector's field time and that of the source representative. Also, if the inspector can obtain all the required data during the inspection, later time-consuming efforts to secure missing data can be avoided. Furthermore, if the inspector has performed his/her homework, the plant personnel are more likely to view the inspector as a professional and to provide the information and cooperation the Agency needs. This section presents guidelines to assist the inspector in conducting a successful inspection.

6.1 PREPARING FOR INSPECTION

When inspecting a ventilation system, the inspector must record the data on site for later use in evaluating compliance practices. The following items will help to ensure that the inspection is complete and that the pertinent information is obtained while the inspector is on site:

Plot Plan

The plot plan should show entrances, major buildings, and the process area to scale and include other appropriate details that provide orientation.

Equipment Drawings

Photographs or sketches of the equipment configuration are useful for reference or comparison. These should show major process and control equipment for easy reference at a later date.

Process Flowsheet and Equipment Checklist

These should provide the inspector with a clear idea of the operating procedures, factors affecting emissions, a listing of necessary data to collect for determining compliance, and data collection methodology.

Before beginning the inspection, the inspector should review the worksheets and process flows with the plant's representative at the plant to assure that the information obtained during the file review is accurate and up to date. This also informs the plant representative of the inspection procedures so that he/she can assist the inspector in collecting information.

6.2 SAFETY CONSIDERATIONS

To aovid injury while conducting a thorough inspection, the inspector must:

- Wear the requisite safety equipment
- Be aware of the safety hazards
- Respect the company's safety procedures
- Never become overconfident

The last point is particularly important. A relatively new inspector or engineer may begin to feel like an "oldtimer" after the first few trips to the plant, and this can be dangerous. For safety reasons, the inspector should make it a practice to stay with the plant escort.

6.2.1 Safety Equipment

For proper fit, the inspector should have his/her own safety equipment. The following equipment is recommended:

- Hard hat
- Safety glasses with side shields or full-cover goggles
- Steel-toed safety shoes
- Fire-resistant pants and jacket
- OSHA-approved respirator (fit-tested)
- Heavy-duty gloves

Although incidents may seem ulikely, when the inspector's attention is often focused on observing emissions and/or operating procedures, he/she can easily become unmindful of potential hazards. Part of the preparation of the observation procedures should be to review the location of the observation point, any required movements, the expected activities in the immediate vicinity, and the availability of an escape route. It is generally advisable to stay as far away from moving equipment as possible.

Unfortunately, a thorough inspection entails some risk. Respect for the hazards, familiarity with operations, and constant concern for safety will minimize the chance of an unpleasant or fatal accident.

6.3 ONSITE COMPANY-INSPECTOR INTERACTION

The success of an inspection depends greatly on the interaction between the inspector and plant representative. Upon arrival at the plant, the inspector should be prepared to discuss the following:

- Authority for the inspection
- Agency organization
- Scope, timing, and organization of the inspection (preferred inspection agenda)
- Treatment of confidential data

It is also important to inquire about the operational status of the equipment to be inspected and the kinds and frequencies of any malfunctions. If equipment is not operating at or near normal conditions, the inspector should note the reasons and when the plant expects it to be operating normally (for followup inspection scheduling).

Before collecting any data, the inspector should observe process operations for a while to become familiar with process variations, wind patterns, plume characteristics, etc.

6.4 INSPECTION PROCEDURES

The inspection and testing of large ventilation systems present some problems not normally associated with smaller ventilation systems. Most of the problems are associated with access to the ductwork and the physical size of the system. In many instances, access limits the measurements that can be taken.

The first criterion in the evaluation of the performance of large systems is to determine whether capture is proper at the source. Poor lighting sometimes limits visual observation of capture capabilities and makes it difficult to evaluate the effectiveness of the system. The second criterion is to determine the integrity of the ductwork to the control equipment.

Again, access conditions may limit this visual inspection to what can be observed from the ground or from a nearby roof. Nevertheless, excessive inleakage is occurring and the need for maintenance should be determinable.

6.4.1 Duct System Inspection

The inspector should have the following before he/she begins the inspection:

- Knowledge of the operations in the process.
- Knowledge of the physical/chemical nature of the process charge materials.
- A layout of the operations showing the controlled and uncontrolled areas.
- A line sketch showing the elevation(s) and layout of the ductwork; the locations of the collector, fan, control equipment; and the flow patterns.
- Layout(s) and sketch(es) of the duct size (length and diameter) showing the main duct, branch or interconnecting ducts, and their respective flow (actual cubic feet per minute); temperature (°F) of actual gas flow; slope ratio for transition points of branch ducting to the main or interconnecting ducts and their respective angles; type and thickness of duct lining; and the location, type and size of blast gates and dampers and how they are controlled.
- Copies of plant tests for velocity, pressure drop, and temperature, and the production or process rate during the test.
- The production or process rate during the inspection.

During the inspection, the inspector should note the condition of the duct (e.g., erosion, corrosion, rusted-through openings), flanges, expansion joints, fits of swing-away joints, etc., all of which affect the workability of a ducting system.

He/she should check to see that any emergency air inlet dampers, such as those located in the duct on electric arc furnace systems, are closed. These are designed to open when temperatures are high, and they should close automatically and stay closed during normal operation.

For inspection purposes, the ventilation system is defined as the ductwork leading from the emission points to the control devices. It is recommended that static pressure taps be made throughout the length of this ductwork to provide data on air inleakage and ductwork plugging, two of the major problems with ventilation systems. Both will change the static pressure and temperature profiles of the ductwork system. If a problem is indicated, the inspector should carefully examine the ventilation system to pinpoint the problem.

Duct blockage, which is characterized by a large increase in static pressure between the blockage and fan, reduces hood face velocity and results in failure of the hooding to capture fugitive emissions. Blockage typically occurs in ductwork bends and cooling loops and may be attributable to insufficient duct velocity (improper duct sizing), sedimentation of dust particles, or excessive cooling of the gas stream (which tends to change the particulate matter into sticky particles that deposit in the duct).

Air inleakage can be a major contributor to excessive cooling of the gas stream. Although air inleakage may not occur at a single point, it is characterized by lower static pressures and lower temperatures downstream of the inleakage points. Excessive inleakage may result in fugitive emissions due to decreased collection efficiency at the emission point and, as noted above, increased potential for duct blockage.

The inspector should check the face velocity and positioning of all fugitive hooding. Improper positioning of hooding or hood damage can reduce capture efficiency. Also, a negative pressure of at least 24 to 49 Pa (0.1 to 0.2 in. H_2O) should be maintained at fugitive and process emission points to accommodate any surges in gas volume and emissions.

6.4.2 Fan Inspection

Lower duct static pressures can also indicate an undersized or underperforming fan. The fan system components should be inspected for wear or corrosion and excessive dust buildup or grease accumulation removed. Fan couplings should be inspected for loose bolts or misalignments. Bearings should be clean and lubricated. Shaft seals should be inspected for leakage. Although a certain amount of leakage is tolerable, excessive leakage may indicate a need for seal replacement.

A major factor to be checked during an inspection is fan vibration. Vibration is a function of fan speed. Normal vibration amplitudes at different fan speeds are as follows:

Speed, rpm	Normal vibration amplitude, in.
400	0.003
800	0.002
1200	0.0013
1800	0.0008
3600	0.0005

Although vibration amplitudes up to 2.5 times the normal value are acceptable, corrective measures are required when this limit is exceeded.

Excessive vibration can result from several things, including material buildup on a blade or bearing wear. Excessive vibration requires immediate attention, as it may quickly lead to catastrophic failure of the fan. An out-of-balance fan wheel rotating at 300 to 700 rpm can fracture the shaft and break through the housing. This can damage other equipment in the area and endanger the welfare of people in the immediate vicinity. The inspector should vacate the area immediately and promptly report any severe vibration to the plant.

Performance tests may be necessary to ensure the proper fan operation after major maintenance, including rebalancing the fan wheel if any repairs were made to rotating components. Using a pitot tube and manometer, an inspector can measure velocity pressures at various points in the duct and calculate the volume flow rate from these readings after correcting them for density at the operating temperature and pressure. Fan horsepower can be calculated from readings of voltage and current supplied to the motor, but it also must be corrected for actual density. The horsepower and static pressure should be plotted on the fan's original characteristic curves for comparison.

Each fan has a set of isolation sleeves, the size of which depends on the fan size. Cracks or tears in the inlet sleeves permit in-draft air to be pulled in by the fan, which reduces suction from the hood. In high-temperature operations, inlet sleeves are made of an asbestos compound. In lower-temperature operations, they are made of neoprene type rubber. Rubber sleeves should never be painted; any type of paint will attack rubber.

Fans are designed to rotate in a given direction. If electrical connections are reversed, the fan will still move air, but not efficiently. Fan rotation direction should be confirmed against design direction.

Fan curves cannot be used directly on a scrubber that has water droplet carryover, as the water droplets represent a mass that the fan has to accelerate. A wet fan indicates that a mist eliminator isn't working properly. This will require the fan to have higher horsepower, and the fan curve will inaccurately project that more gas is going through the system than there actually is. Uncondensed moisture in the gas stream also must be corrected because it changes the density. At some sources, a Fyrite test kit can be used to check the change in oxygen content throughout the system. Severe inleakage will cause a sudden jump in oxygen content. This technique cannot be used for an ambient system.

6.4.3 Gas Flow Check

Gas flow is a useful parameter in evaluating system performance because it provides an indication of the capture velocity at the hood(s). In general, as the quantity of the gas moved through the fan increases, the required horsepower also increases. This increase is reflected by an increase in motor current, which is often measured in a control room. Although motor current sometimes can be measured by a portable clamp-on ammeter, inexperienced personnel should not attempt this kind of measurement, particularly at high operating voltages. The current flow is a useful indicator for all types of systems, regardless of the flow control method (i.e., damper type or speed control). Static pressure across a fan operating at a fixed speed and without inlet spin-vane dampers also will indicate gas flow; gas flow decreases as static pressure or resistance increases.

The use of fan curves is one method of determining gas flow. This requires the measurement of fan rotation speed, gas temperature, and motor horsepower or static pressure across the fan. This method cannot be used if inlet spin vane dampers are used.

The use of a pitot traverse is a more traditional approach to determining gas velocity. In this method (outlined in EPA Reference Methods 1 and 2) a pitot tube is used to measure velocity profile across the duct and the known cross-sectional area of the duct for calculating gas volume through the system. Again, the limiting factor may be access to the ductwork. Long, difficult-to-handle pitot tubes may be required to obtain the measurements in large ducts.

Once gas volume is estimated, the average hood capture velocity may be estimated by comparing the measured gas volume with the cross-sectional area of the hood. This indicates whether the average hood face velocity is adequate to provide the desired capture efficiency. When multiple-hoods are interconnected, the flow in each branch may have to be measured to establish the proper flow balance from each hood. The difficulty with this method is that it provides only an average hood face velocity, which may be substantially different from localized velocities in large ventilation hoods.

Pitot tubes generally will not accurately indicate hood face velocity at the relatively low velocity encountered at most ventilation hoods (250 to 750 fpm). Therefore, other types of instrumentation must be used such as hot-wire and low-pressure gauges. Hand-held instruments such as vane and propeller-type anemometers cannot be used to measure localized face velocity. Hot-wire anemometers may be attached to long probes for measurement of face velocity at various points across the hood face, but the results would have to be corrected for the additional wire length (resistance), which would affect the readings. Again, access would have to be provided for these tests, which could be impractical or unsafe during source operation. A test of this nature, however, would establish minimum and maximum hood face velocities, flow distribution, and average hood face velocity, whereas measurement of the gas volume/hood opening area provides only an average velocity.

6.4.4 Visual Observations

If no measurements are made, inspection is limited to visual observation of the system performance, including hood and duct integrity, the effects of cross-drafts, and operational procedures that may affect the capture efficiency of the ventilation system. Hoods should remain intact and as close to the source as possible, not only to capture emissions, but also to reduce gas volume requirements. Ducts should be checked for damage, which can cause pressure losses, wear, and inleakage. Cross-drafts should be minimal. If the system is not designed to operate with cross-drafts, it may not perform satisfactorily. Wind direction and open doors and windows should be noted.

6.5 OPERATION AND MAINTENANCE (O&M) CONSIDERATIONS

Because the fan systems used in large ventilation systems represent substantial capital outlay and are costly to operate, it is in the best interest of the plants to maintain this equipment in a fashion that lowers the incidence of failures that lead to excessive downtime and extends the useful life of the equipment. The fan system should be considered as including the fan, the motor and drive systems, the inlet and outlet duct systems, and any flow-control dampers used to control the quantity of gas being moved through the ventilation system.

The successful operation and maintenance of the fan system on large industrial ventilation systems does not depend on any one item, but on the proper design and operation of several components. Although some items are more critical than others, all must operate as a unit for the ventilation system to deliver the desired gas volume most efficiently.

For economic reasons, most single fan installations are limited to gas volumes of approximately 1.0 to 1.3 million acfm at nominal static pressure drops. If greater gas volumes are needed than can be economically delivered by a single fan, two or more fans arranged in parallel may be used. This arrangement allows the use of several smaller "off-the-shelf" fans, which are less costly than custom engineered fans; however, when multiple fans are used, care must be taken to see that the ductwork and fans match, particularly when fans of different sizes are used. This approach permits the use of smaller individual fan drive motors and also allows the ventilation system to remain partly operational in the event of the failure of one of the fans or fan drive motor systems.

Although many components in the fan system may be subject to failure, two areas are of the greatest concern: 1) improper balance and the resulting excessive vibration of the fan, and 2) failure of the drive motor system. Either can result in expensive repairs or replacements. Proper design and a preventive maintenance program can reduce the incidence of such failure.

Some drive systems are quite large and require motor sizes in excess of 500 hp (up to as high as 8000). The initial purchase is generally proportional to the motor size. On the other hand, the larger systems tend to be more efficient, and energy savings over the life of the unit may offset the initial cost. Proper matching of motor size with horsepower requirements also will usually increase motor efficiency.

Most drive motors are at their highest efficiency and lowest power factor losses when operated at 80 to 90 percent of their maximum rated load. If motors are always operated at 100 percent of rated load, they may require more maintenance and may not have sufficient reserve for occasional periods when more horsepower is needed. The "service factor" rating of the motor, which usually ranges between 1.0 and 1.15, is an indicator of the sturdiness of the motor and its ability to run at higher than rated load for an extended period of time without experiencing damage.

Because many of these fans are used in systems that control fugitive emissions varying in gas temperature (and gas volume to be captured), the maximum and minimum gas temperatures must be considered in sizing the fan motor. If the fan is sized only for gas conditions at the maximum operating temperatures, problems with motor overload may occur when gas temperature decreases. This results in a denser gas, and more energy is required to move it. Eventually, such continual overloading of the motor may lead to burnout of the motor windings and motor failure if efforts are not made to minimize the problem.

Motor startup is also related to motor overload. The larger the motor size is, the higher the operating voltage required to keep the current flow at a reasonable level. Operating voltage will typically be 440 to 460 for motors having 350 hp or less (but in some cases up to 500 hp). For larger motors, operating voltages generally increase to 2130 or 4260. When a fan motor is started "across the line," however, it can draw 6 to 7 times its normal operating current (regardless of voltage) during acceleration of the fan wheel to its normal rotation speed. Although this current surge is of short duration and diminishes as the fan approaches its running speed, it could damage the motor on startup with a cold gas stream and cause circuit breakers to trip, both at the fan and in other areas of the plant that might be affected if the surge of current into the fan circuit were to reduce operating voltages to unacceptably low voltages and cause undervoltage trips.

Two methods that can be used to reduce the possibility of circuit breaker trips or damage to the motor upon startup are reduced-voltage starts and closed-damper starts. Reduced-voltage starts allow the fan wheel to accelerate to a portion of its rotation speed, and then full voltage is applied to accelerate the fan wheel and gas to full speed. Although the current will still surge upon startup, the levels of surge should be more acceptable, particularly if other circuits are involved. Closed-damper starts allow the motor to accelerate the fan wheel without simultaneously moving the gas; much of the current surge results from accelerated gas flow through the system. Opening the dampers gradually to allow gas to flow after the fan wheel is rotating permits the flow of current to be controlled.

A third method is to control the rotation speed. This method is similar to the reduced-voltage start. The primary difference is that this method uses a variable-speed motor or a transmission coupling, which allows variable fan rotation speeds. A gradual increase in rotation speed controls the current surge.

Excessively high currents in the motor create heat in the windings, which can destroy the winding insulation and result in the loss of windings because of short-circuiting. Heat buildup in the motor can be a major reason for motor failure. Other reasons for heat buildup in the motor include improper or restricted ventilation due to the location of the motor or excessive dust buildup. Motors must be kept clean to maintain the flow of cooling air through them. Additional cooling considerations may be necessary if fans are located where ambient temperatures are high. Normally, solid-state controls for variable-speed motors also must be protected from high temperatures.

Transmission of the motor energy to the fan shaft is usually accomplished by direct drive, by V-belts, or through a variable-speed transmission. The belts must be tensioned properly and kept free of grease or oil to prevent slippage and belt damage. The fluid levels of fluid-drive transmissions for variable-speed operation must be maintained (and possibly cooled) for reliable operation.

Most fan shafts are supported by bearings at the fan housing and at the drive connections. Worn bearings can cause excessive fan vibration and increase energy costs. Bearings for smaller fans can be installed with grease seals and a grease fitting for routine lubrication. The bearing lubrication should be checked at least daily to ensure that it is adequate.

On fans operating with high-temperature gas streams, it is particularly important to keep the bearings cool to prevent breakdown of the lubricant. Heat can be transmitted through the shaft to the bearing assembly.

Although most fans are designed with some inleakage of air around the shaft to the fan housing, this may not be enough for adequate cooling. In this case, heat fins may be installed on the fan shaft where it exits from the fan. These fins rotate with the shaft and provide extra cooling surface by conducting heat away from the bearings. In situations where this approach is inadequate, a continuous lubrication system that utilizes circulating oil can be applied. This system, which consists of a pump, circulation lines, a filter, and a cooling system (usually water), simultaneously provides bearings with continuous lubrication and cooling. More complex than simple grease lubrication, this system has the disadvantage of requiring continuous operation.

On large fan systems, the use of vibration monitoring equipment can prevent continued operation of a fan that is unbalanced or has worn bearings. If a severely vibrating fan is allowed to continue operating, a "fan explosion" could occur when the fan is no longer able to withstand the stresses and simply comes apart during operation. This can be extremely dangerous to personnel working in the area and also costly to repair.

All fan systems are equipped with inspection hatches and the fans should be inspected through these hatches at least annually to evaluate the severity of fan wheel wear. Generally, evidence of fan blade wear appears on the edges of the fan wheel, on the side opposite the inlet. The chance of fan wear and blade buildup is greater on fans that are installed in the gas stream prior to the air pollution control equipment than on fans placed downstream of the control equipment, and they should be checked more frequently for wear and vibration. When blades are replaced, the fan usually must be rebalanced.

Particulate buildup on fan blades can lead to improper fan balance. Because the buildup on fan wheels tends to be relatively uniform, it does not affect the fan wheel balanced until it flakes off, at which time a substantial change in fan balance can occur. This, in turn, leads to fan vibration and bearing wear, and can eventually cause fan wheel failure. This buildup problem can be severe when the dust is sticky or oily or if the fan follows a wet scrubber, in which case wet gas conditions enhance the buildup problems. (Water can be applied to a fan that is used after the scrubber to prevent particulate buildup.) The higher the rotation speed of fans, the more susceptible they are to the severe inbalance problems of particulate buildup.

Fans with linings should be inspected as often or more often than other fans to ascertain that the lining is still intact. Care should be taken not to damage the lining during inspection. Because fans are most often lined for corrosion resistance, these linings must be kept in good condition to prevent excessive fan wheel or housing corrosion. The key to the successful application of lining is good surface preparation, which seems to be as much an art as a science.

Another area of concern is the delivery of proper gas flow through the ventilation system. This flow control is usually accomplished through the use of dampers or by controlling the rotation of the fan speed. The latter method is the most energy-efficient. The quantity of gas delivered by a fan is proportional to the change in rotation speed, but the horsepower required changes by a cubic relationship. Thus, very small changes in rotation speed can result in significant changes in horsepower. The initial cost of the speed control equipment will offset the energy savings somewhat.

The next most efficient method of controlling gas flow is the use of inlet spin vane dampers. As the vanes close, gas is spun in the direction of the fan wheel rotation. The quantity of gas delivered by the fan is a function of the difference in inlet and outlet gas velocity. Because the gas is spinning in the direction of rotation, the difference in velocity is smaller and the quantity of gas delivered is smaller with no waste in the static pressure developed by the fan. These dampers should be checked periodically to guard against improper functioning and excessive wear of the vanes.

The least efficient but simplest method of controlling gas flow involves the use of blade-type dampers. These dampers place additional resistance (variable) into the system to control the gas flow. Even at reduced flow rates, the fan still delivers a high static pressure and thus requires more energy to move the reduced gas volume. This method is also the simplest to maintain unless excessive wear occurs.

Air inleakage is one reason why systems may not perform as designed. Air can flow into the system wherever a leak occurs, e.g., through a hole in the ductwork, an expansion joint, or open branches that should be closed. The fan will pull gas through the point of least resistance, and this can leave the ventilation system with inadequate draft. Any damage should be repaired before it leads to inadequate control of emissions, corrosion (leading to increased air inleakage), and higher energy costs for transporting unwanted dilution air though the system.

Ductwork design also affects how much gas the system will deliver. The shortest, straightest duct is the most desirable for the design, operation, and maintainance of the system. On large systems, rectangular ducts are often easier to install than round ones because it is easier to weld large flat sections of steel; however, particulate matter has a tendency to build up in the corners of rectangular ducts and close off usable area. For this reason rectangular ducts may require periodic cleanout. On the other hand, some fans (e.g., double-inlet centrifugal fans) are more adaptable to rectangular ducts.

Improper design of the inlet design of the ductwork can cause a swirling gas flow to the fan, which creates an effect similar to that obtained from inlet spin vane dampers. This can cause reduced gas flow and inadequate capture by the ventilation system. This is particularly true if smooth transitions are not provided to the fan, in which case flow-straightening vanes or redesign of the ductwork may be required.

SECTION 7

TOTAL FURNACE ENCLOSURES

This section discusses total furnace enclosures in the metallurgical industries. It is assumed that the reader is generally familiar with the metallurgical processes that use these enclosures. The basic ventilation principles outlined in previous sections are valid for these special applications.

7.1 ELECTRIC ARC FURNACES

The recent trend in electric arc furnace (EAF) emission control is to totally enclose the furnace operations. This system allows the collection of both primary and secondary emissions from EAF operations. Several smaller furnaces (<100 tons) have installed total furnace enclosures. The essential features of these enclosures are sliding doors that create access for the crane and scrap charging bucket and an air curtain to block the escape of fumes from the roof when the roof doors are parted for crane cable access during charging. The advantages of total furnace enclosure are:

- Effective fume capture.
- Low volumes of air handled as emissions collected at the source.
- Capture of both primary and secondary emissions.
- Access for furnace maintenance.
- Lower noise levels outside the enclosure
- Lower capital and operating costs.
- Better working environment and lower roof temperature in the EAF shop.
- Minimal effect of cross winds because the entire operation is enclosed.

- Lower maintenance of cranes and other equipment because of reduced dust-fall within the shop.

Figure 32 illustrates a typical furnace enclosure.

7.1.1 Description

A typical enclosure should have adequate space around the furnace to provide a suitable working area and reduce the effect of heat on the enclosure structure and furnace components. The enclosure structure is generally lined with hi-rib aluminized sheeting and joints sealed with closure strips. Bi-parting vertical doors in front of the furnace are controlled by air cylinders to allow the overhead crane to enter the enclosure. A rectangular roof slot provides clearance for crane cables when the crane is operated within the enclosure. This slot is closed by roof doors that are pneumatically operated. An air curtain under the roof slot area contains and guides the emissions to the pickup hood located opposite the air curtain nozzles. The air curtain fan is usually located on the roof of the furnace transformer room. Removable roof panels above the roof doors allow access for furnace maintenance. The furnace operations, air curtain, and pickup hood can be observed safely from the control room. The control room usually has a full glass window that forms a part of the enclosure wall. Doors to the enclosure allow access to the furnace for oxygen lancing and for taking metal and slag samples.

To charge the furnace, the crane operator positions the charging bucket in front of the enclosure and aligns the crane cable in front of the roof slot. The furnace operator swings the furnace roof aside and opens the front vertical bi-parting doors and the roof doors of the enclosure. The crane operator then brings the charging bucket into the enclosure. At ground level, a furnace operator guides the crane operator in positioning the charge bucket directly over the furnace. The crane operator dumps the scrap by opening the bottom of the bucket and then reverses the crane out of the enclosure. The control room operator closes the vertical and roof doors and swings the furnace roof in position to seal the furnace. The melting operation commences after the furnace roof is in position. During the entire period when the bi-parting doors and roof doors are open, the air curtain is in operation along with the enclosure exhaust. During melting,

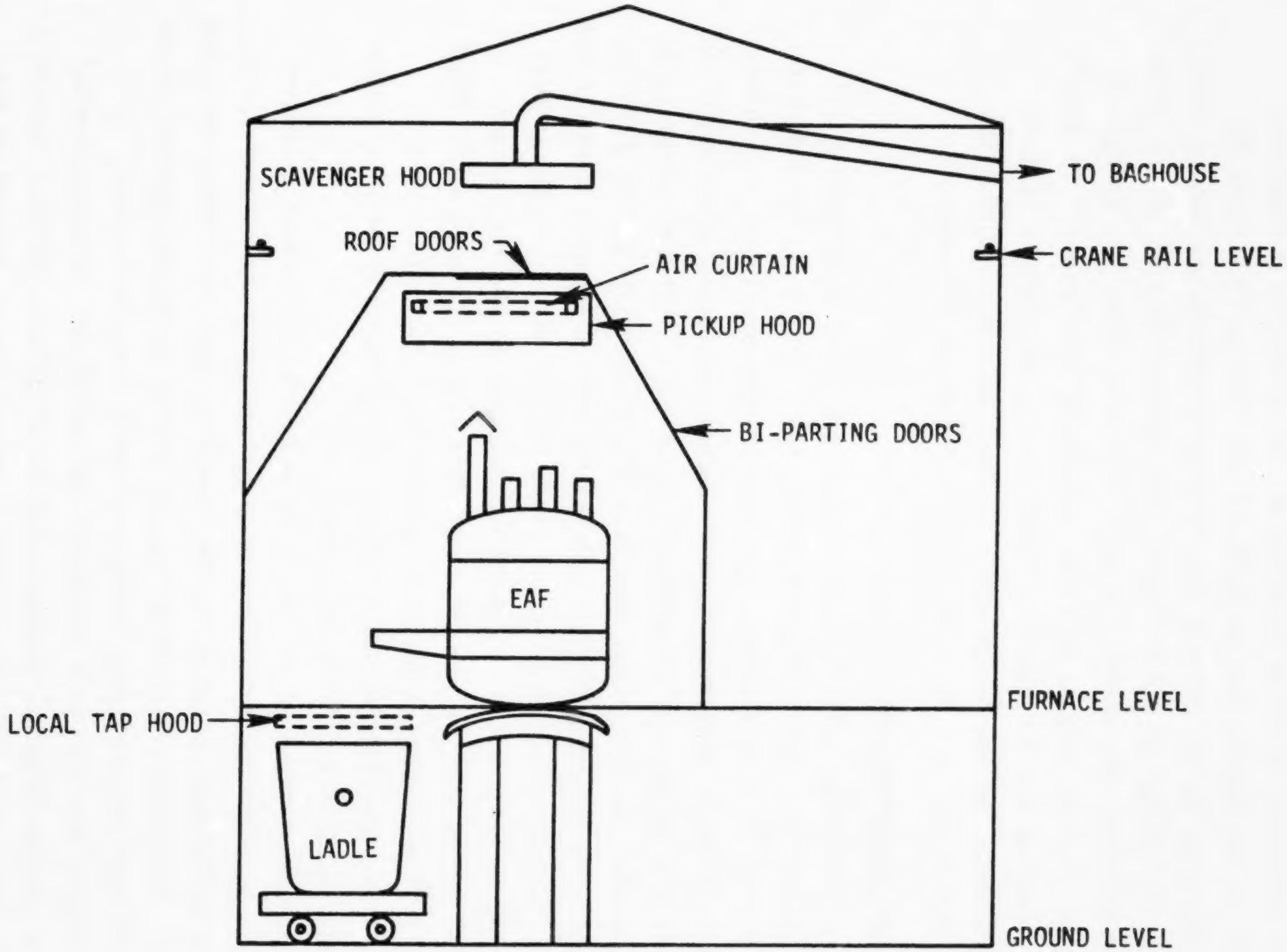

SCAVENGER HOOD
TO BAGHOUSE
ROOF DOORS
CRANE RAIL LEVEL
AIR CURTAIN
PICKUP HOOD
BI-PARTING DOORS
EAF
FURNACE LEVEL
LOCAL TAP HOOD
LADLE
GROUND LEVEL

Figure 32. Typical furnace enclosure.
(Not to scale)

oxygen lancing, deslagging, and sampling, the doors are closed and the enclosure is exhausted to a gas cleaning device, generally a baghouse.

A stack in the fourth hole position of the furnace roof leads the emissions generated during melting and refining closer to the exhaust hood. Depending on the design of the enclosure, the furnace tapping occurs either within or outside the enclosure. In most cases a transfer car brings the teeming ladle into the tapping position underneath the furnace. If tapping takes place outside the enclosure, a local hood is provided to capture tapping emissions. In the design where tapping takes place within the enclosure, the enclosure exhaust evacuates the tapping emissions.

7.1.2 Design Considerations

Charging, melting, refining, and tapping emissions occur during the operation of the EAF. The maximum emissions (85 - 90%) occur during meltdown or refining.[18] Direct shell evacuation or side draft hoods adequately control these primary emissions. Several techniques including canopy hoods and building evacuation control secondary emissions. Although canopy hoods have few operating restrictions, they have the disadvantages of high volume requirements, high capital, and high operating costs. Crosswinds within the shop affect capture efficiency. Local hoods have provided limited success in collecting tapping fumes. A compromise solution to the high volumes required for canopy hood systems is a modified canopy approach where fixed curtain walls form a partial enclosure around the furnace to act as a chimney to direct charging emissions to the canopy hood, while local hoods are utilized to control tapping emissions.

The ultimate control of furnace emissions involves furnace enclosure technology. On small electric furnaces, complete emissions control can be achieved by collecting emissions at the source by adopting furnace enclosure technology. Emissions generated during all phases of the EAF operations can be withdrawn from the enclosure using relatively low flow volumes. If emission volumes are high as a result of the use of UHP furnaces, oxyfuel burners, or larger furnaces, however, additional primary controls such as a DSE (fourth hole) system may be required. The approximate volume of air required for a furnace enclosure in a small furnace will be equivalent to the volume required by a side-draft hood system; a larger furnace will require an amount equivalent to the volume required by a DSE system.[18] The actual air flow required will depend on many factors such as furnace size, oxygen lancing rate, exhaust-duct position, scrap cleanliness, furnace power, types of steel produced, operating practices, and enclosure volume.

7.1.3 Retrofit Installations

Because each retrofit installation is unique, enclosures must be tailored to meet individual requirements, furnace practices, and shop layout. Examples of two retrofit EAF enclosures in the United States are found at the North Star Steel Company and Birdsboro Corporation.

North Star Steel Company--

This company has two 60-ton furnaces that are controlled by DSE and a roof canopy system. The capture efficiency of the canopy system was low during charging and tapping because of the flat design of the roof canopy and insufficient extraction volume. Each furnace was retrofitted with an enclosure. The furnace enclosure is a steel structure clad with ribbed aluminized sheets and sealed. Mechanical bi-parting side doors and roof doors allow entry of the crane for charging. The bi-parting vertical doors are closed during charging and an air curtain seals the roof. The entire available flow is used to extract the charging emissions through a high-level evacuation duct. Emissions from melting, oxygen lancing, and slagging operations are controlled by a DSE system. During tapping, mechanical bi-parting doors on the tapside of the enclosure are opened to permit entry of the tap ladle. These doors have a top horizontal roof with a slot for cables holding the tap ladle. Once the ladle is positioned, the bi-parting doors close to contain the fumes. During tapping, fume is evacuated by a high-level duct. On the charge side, the mechanical roof door, the roof section complete with air curtain, and the charge section evacuation duct can all be mechanically wheeled aside for maintenance operations such as replacement of furnace parts and removal of the shell and roof. Salient data on this plant are as follows:[19]

Furnaces:	Two: 15-ft diameter; 60-ton capacity each
Transformer:	30,000 to 33,000 kVA

Meltdown rate: 55 tons/hour

O_2 consumption: 300 ft^3/ton

Approximate enclosure size: 40 x 36 x 31 ft high

Extraction volume: 170,000 acfm per enclosure

Air curtain volume: 6000 acfm

Figure 33 shows the enclosure at the North Star Steel Company.

Birdsboro Corporation--

This new facility initially designed with a side-draft hood for primary emission control had to be redesigned to control both primary and secondary emissions. The furnace enclosure design allowed an acceptable solution with an exhaust volume of 75,000 acfm for a 40-ton furnace. Tapping and charging are carried out from the same side. A mechanical hood car moves away from the enclosure to provide an opening for the scrap bucket and tap ladle. Three cylinder-operated movable roof doors seal the crane cable entry slot. These doors are linked to dampers in the air-curtain duct system. The air curtains seal the opening when the doors are opened for charging or tapping. Overhead removable panels at the enclosure roof level aid maintenance operations. In the initial startup, the enclosure was not effective during lancing. Measures adopted to overcome this problem were better sealing of the furnace enclosure, upgrading the air curtain fan, adjusting air nozzles to deflect the lancing emissions into the extraction duct, and installing a deflector plate. Salient data on this shop are as follows:[19]

Furnaces: One: 13-ft, 6 in. diameter; 40 tons capacity

Transformer: 10,000 kVA

Maximum meltdown rate: 18.6 tons/hour

Oxygen lancing: 250 scfm (design)

Enclosure size (approximate): 36 x 30 x 28 ft high

Extraction volume: 75,000 acfm (design); 81,600 acfm operating

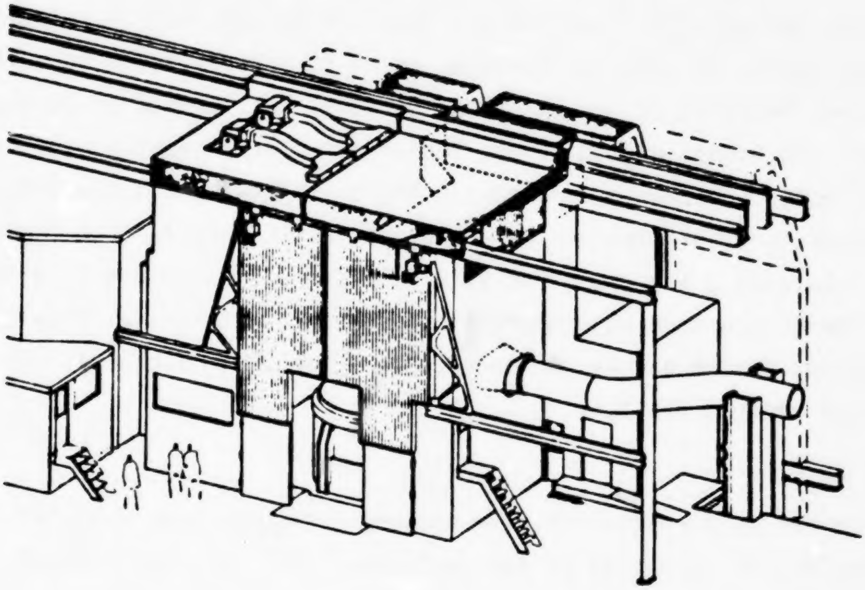


Figure 33. Furnace enclosure at North Star Steel Company.

Temperature: Normal 150°F; Maximum 275°F

Air curtain volume: 9,900 acfm (design) increased to 11,700 acfm.

Figure 34 shows the furnace enclosure at Birdsboro Corporation.

7.1.4 New Installations

Some of the new EAF installations required to meet NSPS have adopted furnace enclosures for smaller furnaces (<100 tons). Table 10 is a list of the essential features of new EAF shops designed for furnace enclosures. Plant 3 has additional pickup points including a local tapping hood, an additional hood to evacuate tundish lancing emissions, and a scavenging hood located above the roof slot of the enclosure, all connected to a common baghouse. Balancing the draft for such multiple pickup points is difficult, but adjustments are made with operating experience. A typical flow balance during various phases of the EAF operations is shown in Table 11. Dampers on each hood control the flow.

7.1.5 Inspection

To achieve good evacuation, the enclosure should be well sealed. This will depend on the integrity of the enclosure. The inspector should observe and note the general condition of the enclosure walls in regard to any loose sealing strips, misalignment of the bi-parting doors (which creates large gaps), holes in the panels, damaged or warped panels, open access doors, and any other openings that reduce the capture efficiency of the enclosure. During scrap charging and at other periods when the air curtain is operated, the inspector should observe the air curtain as it diverts the thermally-driven plume into the pickup hood. Improperly placed air nozzles in the air curtain may not completely seal the roof door area and thus allow emissions to escape. The air curtain operation can be safely observed from the control room. The performance of the enclosure can be assessed by simultaneous observations by two inspectors, one inside the control room and the other outside the enclosure, during all phases of a heat cycle. The inspector in the control room should observe and note the fit and closure of the roof doors, and the buildup of emissions within the enclosure and how effectively they are evacuated at various stages of the furnace operation. The inspector

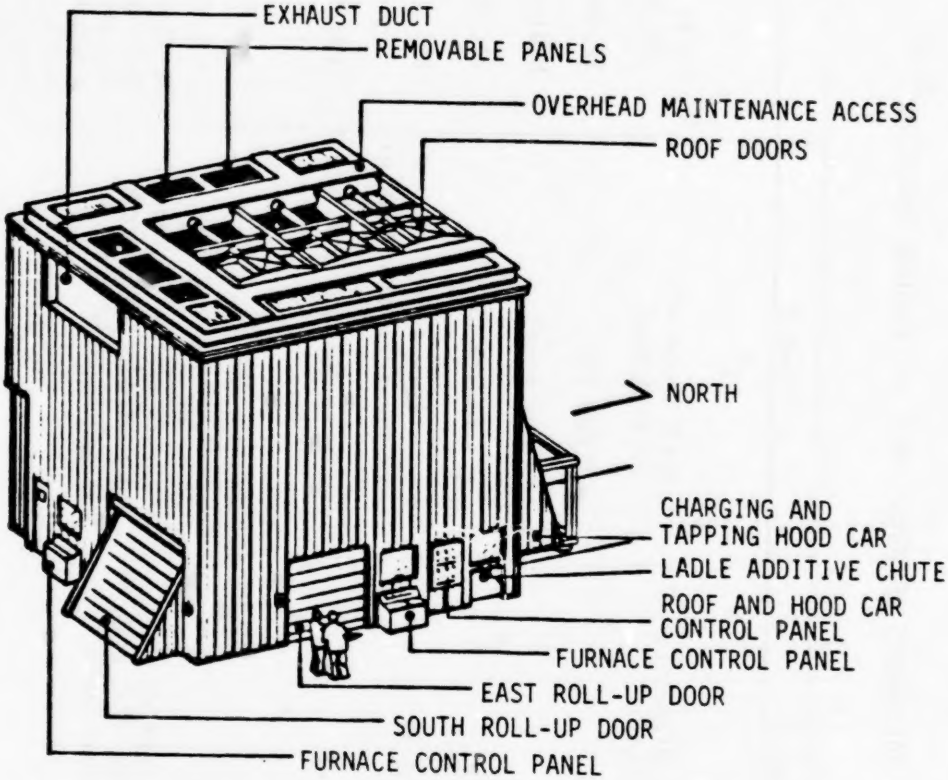

EXHAUST DUCT
REMOVABLE PANELS
OVERHEAD MAINTENANCE ACCESS
ROOF DOORS
NORTH
CHARGING AND TAPPING HOOD CAR
LADLE ADDITIVE CHUTE
ROOF AND HOOD CAR CONTROL PANEL
FURNACE CONTROL PANEL
EAST ROLL-UP DOOR
SOUTH ROLL-UP DOOR
FURNACE CONTROL PANEL

Figure 34. Furnace enclosure at Birdsboro Corporation.

TABLE 10. DATA ON EAF PLANTS DESIGNED FOR TOTAL FURNACE ENCLOSURE[20,21]

	Plant 1	Plant 2	Plant 3
Number of furnaces	1	2[a]	1
Furnace size	12-ft,-6 in diameter; 28-32 tons/heat	16-ft diameter; 60 tons/heat	15-ft diameter; 55 tons/heat
Transformer	16,800 kVA	33,600 kVA	17,000 kVA
Type of steel made	Specialty	Carbon steel	Medium-carbon steels
Meltdown rate	≅20 tons/hour	≅72 tons/hour	≅16 tons/hour
Approximate enclosure size	42 x 51 x 35 ft	34 x 53 x 44 ft (curved at the top, dome shaped)	Not available
Extraction volume	150,000 acfm	175 to 200,000 acfm at 150°F	150,000 acfm at 130°F
Air curtains at the roof slot	Yes	Yes	Yes
Other features	Vertical bi-parting doors and roof slot with doors for crane cable access.	Vertical bi-parting doors and roof slot with doors for crane cable access.	Vertical bi-parting doors and roof slot with doors for crane cable access. A separate tapping hood and a scavenging hood above the crane slot, evacuated to the same baghouse.

[a] Each furnace has a separate enclosure.

TABLE 11. FLOW BALANCING OF A TYPICAL FURNACE ENCLOSURE WITH ADDITIONAL PICKUP HOODS, CONNECTED TO A COMMON BAGHOUSE (150,000 acfm)

Operating Mode	Roof scavenging hood	Furnace enclosure pickup hood	Local tapping hood	Lancing hood
Furnace charging	73,000 acfm	77,000 acfm	Closed	Closed
Melting and refining	73,000 acfm	77,000 acfm	Closed	Closed
Tapping	73,000 acfm	22,000 acfm	55,000 acfm	Closed
Lancing at Lancing Station	43,000 acfm	77,000 acfm	Closed	30,000 acfm

NOTE: Lancing hood cannot be operated during charging or when air curtain is operated, and during tapping. Roof scavenging hood can evacuate at a higher rate than 73,000 acfm, if needed.

outside the enclosure should observe and note the shop layout, any leakage of emissions from the enclosure, crossdrafts inside the shop, emissions fromancillary operations, additional pickup points, and the overall flow balance under different operating modes. Although crosswinds generally do not affect the performance of an enclosure, emissions that build up near the shop roof above the enclosure will escape the building. If a separate scavenger hood is provided above the enclosure, its capture efficiency should be observed.

7.2 BASIC OXYGEN FURNACES

Primary emissions from basic oxygen furnaces are captured in specially designed hoods. The hood is erected above the mouth of the vessel to control the primary emissions when the vessel is in the vertical position. The primary hood is not as effective, however, when the vessel is tilted for various operations such as charging, sampling, tapping, and deslagging. Additional hoods or enclosures are required to control these secondary emissions. The trend is toward total enclosures.

7.2.1 Description

The vessel is enclosed on all sides, with a door or bi-parting doors in the front to facilitate scrap and hot metal charging. Since these operations occur at and above the vessel, natural convection will permit a plume of hot dusty gases to escape during charging. To reduce this possibility, charging should occur as close to the vessel as possible and under the hood. A separate charging hood is provided within the enclosure for capturing the charging emissions. The enclosure can extend partially or completely at the tap side. Tapping is carried out at and below the level of the vessel, and the hot dusty gases have a tendency to escape in the natural draft induced by the process heat. In the newer designs, a permanent tap hood is installed at the back of the enclosure. Figure 35 shows a typical arrangement of a BOF vessel enclosure. The furnace enclosure extends below the charging floor, and the only openings are for the ladle car. These openings can be effectively reduced by the addition of vertical shields on the end of the ladle car.

In the bottom blown BOF (Q-BOP), oxygen is blown through tuyeres. During charging and turndown, gas (either oxygen or nitrogen) must be blown through the tuyeres to prevent liquid metal, slag, or solids from entering and clogging the tuyeres. This generates heavy emissions and makes capture of the secondary emissions more difficult; hence, invariably all the Q-BOP furnaces are completely enclosed.

7.2.2 Gas Cleaning Systems

The gas cleaning system employed to control secondary emissions may be an extension of the primary control system. One hood designed to collect charging emissions and another for collecting tapping emissions could be ducted to the same primary gas cleaning system. The gas flow would be adjusted for the different demands of the heat cycle. Another alternative is to duct the charging and tapping hoods in the furnace enclosure to a secondary control unit, generally a bag filter. Figure 36 indicates a schematic of the secondary control system at Kaiser Steel in Fontana, California.

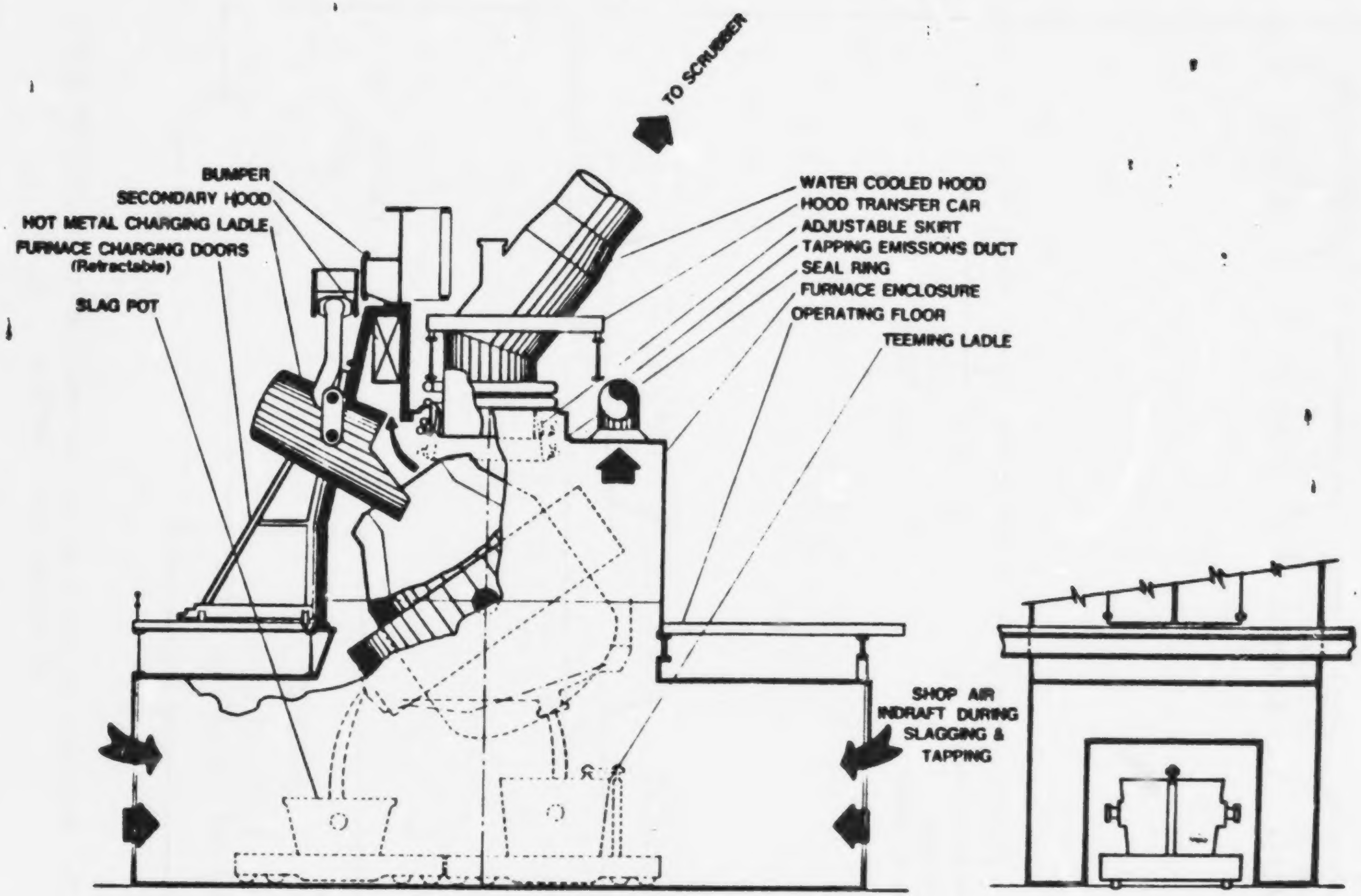

TO SCRUBBER
BUMPER
SECONDARY HOOD
HOT METAL CHARGING LADLE
FURNACE CHARGING DOORS (Retractable)
SLAG POT
WATER COOLED HOOD
HOOD TRANSFER CAR
ADJUSTABLE SKIRT
TAPPING EMISSIONS DUCT
SEAL RING
FURNACE ENCLOSURE
OPERATING FLOOR
TEEMING LADLE
SHOP AIR INDRAFT DURING SLAGGING & TAPPING

Figure 35. Typical BOF furnace enclosure.

Courtesy: Pennsylvania Engineering Corporation.

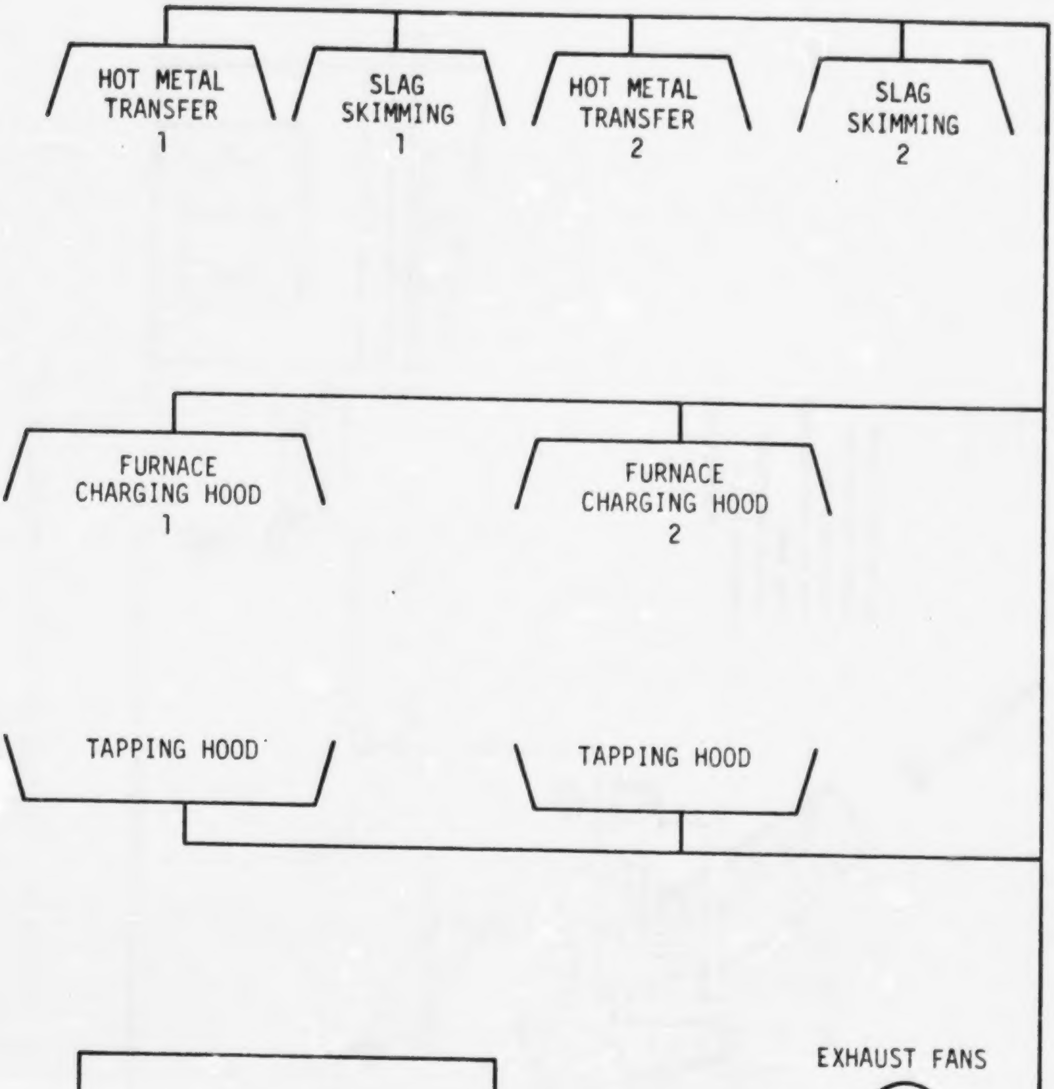

HOT METAL TRANSFER 1
SLAG SKIMMING 1
HOT METAL TRANSFER 2
SLAG SKIMMING 2
FURNACE CHARGING HOOD 1
FURNACE CHARGING HOOD 2
TAPPING HOOD
TAPPING HOOD
EXHAUST FANS
BAGHOUSE
1,020,000 m3/hr
MAXIMUM TEMPERATURE 230°C

Figure 36. Schematic of basic oxygen secondary emission-control system of Kaiser Steel-Fontana.

A common baghouse with a 1,020,000-m³/h (600,000 acfm) design flow serves not only the charging and tapping hoods within the furnace enclosure but other areas such as the hot metal transfer and slag skimming stations.

In the closed-hood system that intermittently handles a flammable gas (CO), use of the primary system to control the secondary sources could result in an explosion.

At the Q-BOP plant of Republic Steel in Chicago, which has a suppressed combustion or closed-hood system, charging emissions are safely collected by diverting the emissions to the primary system of the nonoperating vessel. The shop has two vessels, and one furnace is operated at a time. Figure 37 shows the schematic arrangement of the gas cleaning operation. When both gas cleaning system fans are operated, a flow rate of 634,000 m³/h (373,000 acfm) is available at the charging hood during hot metal charging.[22] Fumes captured in the charging hood bypass the quencher and pass directly to the venturi scrubber. The design pressure drop of the venturi during furnace charging is 218 cm (56 inches) water column.[22]

Table 12 lists the BOF/Q-BOP shops that utilize the furnace enclosure for emission control.

7.2.3 Inspection

The integrity of the furnace enclosure is important in obtaining good evacuation within the enclosure. The inspector should observe and note the condition of the bi-parting doors to detect any misalignments or warpage that leads to large gaps between the doors. The inspector should also check for gaps between the water-cooled sections of the duct work and inspect the condition of the pickup hoods within the enclosure and all the associated ductwork leading to the cleaning unit.

During scrap charging, the vessel tilt angle should be estimated along with the effective capture efficiency of charging emissions in the secondary charging hood. During hot metal charging the tilt angle and the pouring rate of hot metal play an important part in the effective capture of the secondary emissions. The primary hood capture and enclosure evacuation should be observed during slopping. In top-blown furnaces, lanse hole emissions or fume rollout (puffing) from the primary hood indicate insufficient draft at the primary hood. This will increase the load on the secondary controls within the enclosure.

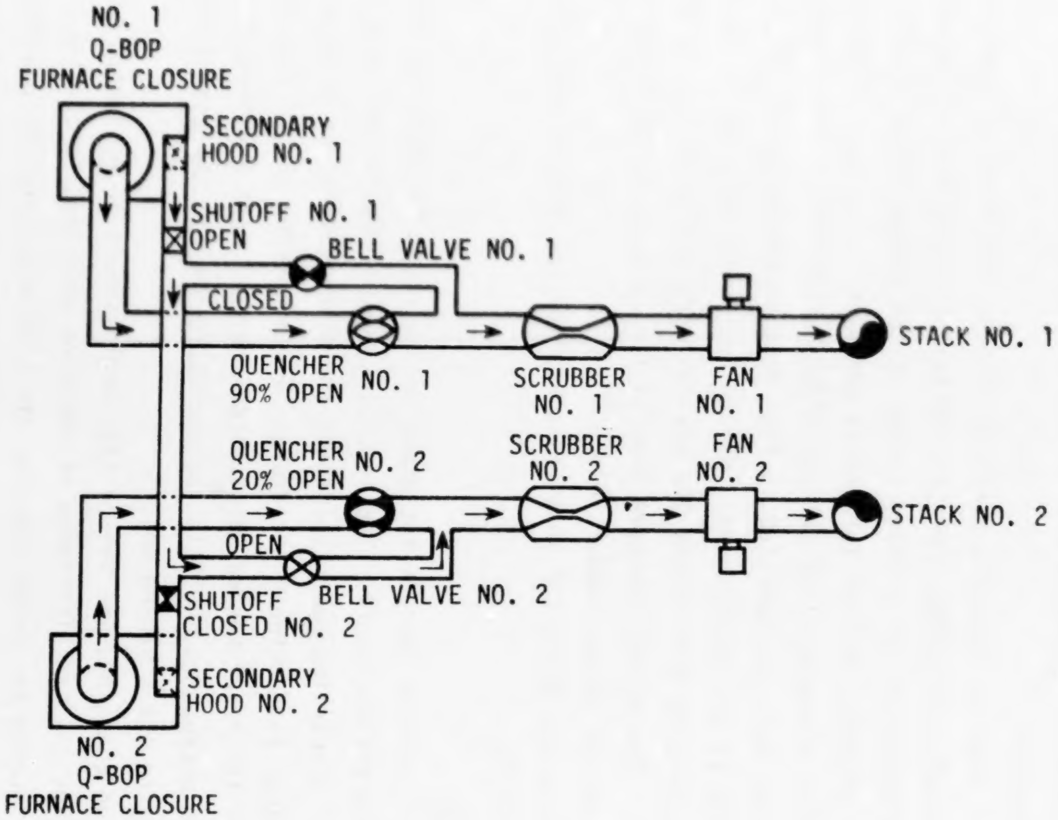

NO. 1
Q-BOP
FURNACE CLOSURE
SECONDARY
HOOD NO. 1
SHUTOFF NO. 1
OPEN
BELL VALVE NO. 1
CLOSED
QUENCHER
90% OPEN
NO. 1
SCRUBBER
NO. 1
FAN
NO. 1
STACK NO. 1
QUENCHER
20% OPEN
NO. 2
SCRUBBER
NO. 2
FAN
NO. 2
STACK NO. 2
OPEN
BELL VALVE NO. 2
SHUTOFF
CLOSED NO. 2
SECONDARY
HOOD NO. 2
NO. 2
Q-BOP
FURNACE CLOSURE

Figure 37. Schematic of Q-BOP secondary (charging) emissions control system of Republic Steel, Chicago.

TABLE 12. BOF/Q-BOP SHOPS UTILIZING FURNACE ENCLOSURES[a]

Company and location	Type of enclosure	Type of gas cleaning	Startup date
Republic Steel Corp. Chicago, Illinois	4-sided enclosure with mechanized biparting charging doors	High-energy scrubber (420,000 acfm at 143°F)	1976-77
Bethlehem Steel Corp. Burns Harbor, Ind.	4-sided enclosure with mechanized biparting charging doors	High-energy scrubber (350,000 acfm at 145°F)	1977
CF&I Steel Corp. Pueblo, Colorado	3-sided enclosure without charging doors	Fabric filter (540,000 acfm at 275°F)	1978
Kaiser Steel Corp. Fontana, California	4-sided enclosure with single mechanized charging door	Fabric filter (500,000 acfm at 600°F)	1978
Granite City Steel Div. Granite City, Illinois	4-sided enclosure with single mechanized charging door	Electrostatic precipitator (900,000 acfm at 550°F)	1980
Rouge Steel Co. Dearborn, Michigan	3-sided enclosure without charging doors	Electrostatic precipitator (1,050,000 acfm at 500°F)	1981
Sharon Steel Corp. Farrell, Pennsylvania	4-sided enclosure with single mechanized charging door	High-energy scrubber (320,000 acfm at 168°F)	1982

[a] Source: Pennsylvania Engineering Corporation.

The inspector should observe and note the shop layout, gas cleaning equipment, and multiple-pickup points connected to the same gas cleaning source and become familiar with the flow-balancing scheme under different operating modes. This is very important as each plant is unique and equipment operation varies from plant to plant.

SECTION 8

SPECIAL APPLICATIONS

The basic principles of ventilation outlined in previous sections are valid for all applications. This section discusses some additional considerations of interest in common control systems in the metallurgical industries. It is assumed that the reader is already generally familiar with the processes involved.

8.1 COKE OVEN SHEDS

8.1.1 Description

The coke oven shed for control of coke oven pushing emissions is a special kind of hooding. Many variations in configuration are possible (Figure 38) but all are designed to meet three basic objectives:

- To contain excessive emissions during the brief (45-second) period of a single push; cooling and exhausting of these emissions take place over a longer period.
- To arrest the upward travel of the plume and to roll the plume within the shed; heavier particles fall out to the ground.
- To contain all of the emissions from the coke side of the battery, including door emissions, coke spillage emissions, and hot car emissions.

Shed control systems require ducting, fans, and a control device, usually a fabric filter. This discussion is limited to the ventilation principles and O&M considerations associated with the shed itself.

Figure 39 presents a general illustration of the thermal expansion of hot gases from a push that the shed must accommodate. In the original shed designs, a duct ran along the length of the shed, and the exhaust suction was distributed along the entire length. These sheds required high flow rates (300,000 acfm or more) to achieve reasonable face velocity. Newer designs

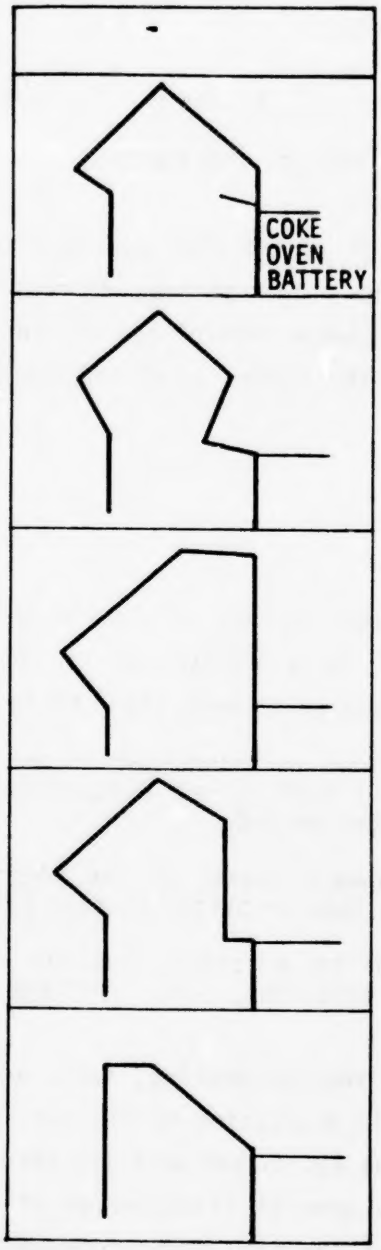

COKE
OVEN
BATTERY

Figure 38. Various shed configurations.

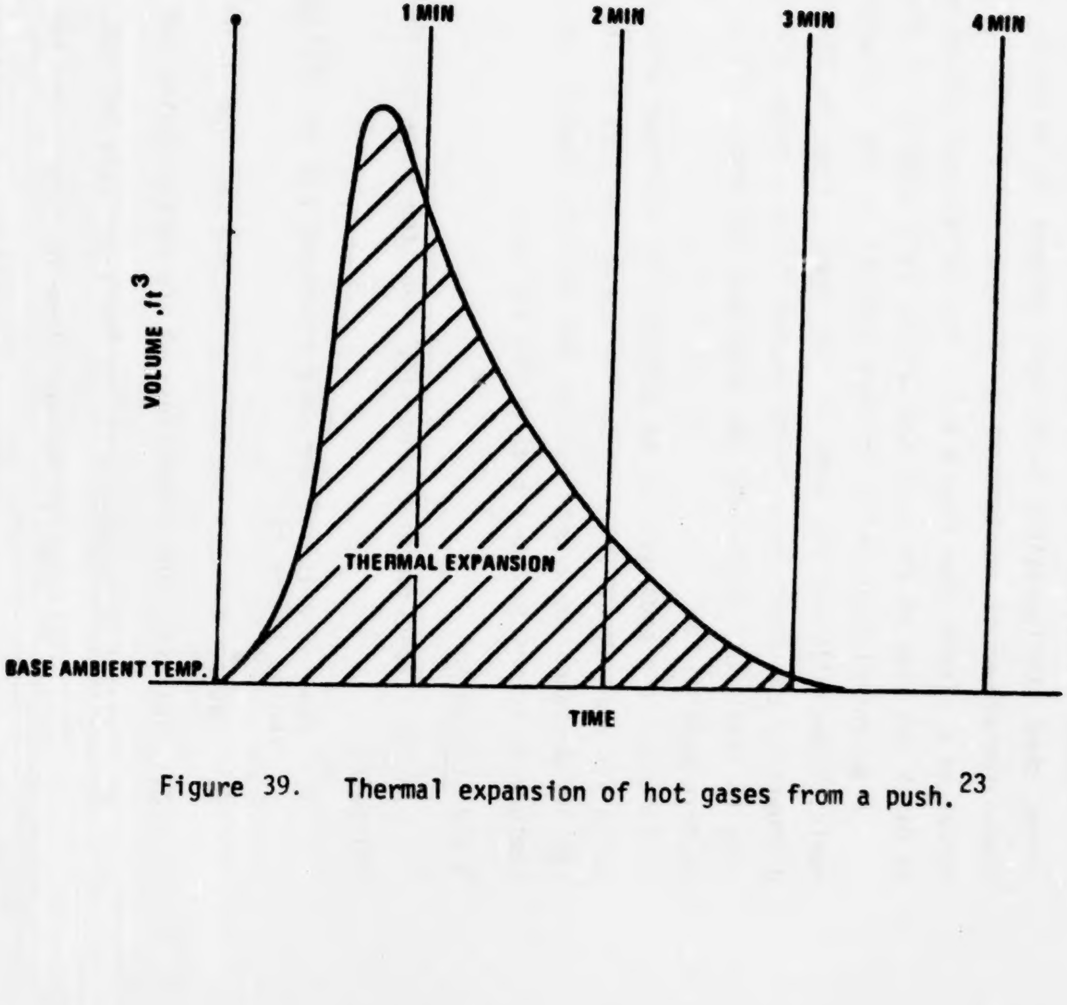


Figure 39. Thermal expansion of hot gases from a push.[23]

entail a partition approach in which dampers are used to concentrate the suction in the segment of the shed where the push is occurring. In either design, an inherent design principle is the holding of it within the large confines of the shed until it can be fully exhausted. The design relies on the theory of fully exhausting a given push before the next push occurs.

Because containment of the plume is an important factor, several different shed configurations have been offered to maximize containment. A common feature of all designs is an enlarged upper portion of the shed, which serves as a plenum (see Figure 38). The containment volume must be designed to match the size of the push and pushing rate expected at the battery.

A potential weak point in shed control is the influence of wind currents, especially at the ends of the shed. Even in the absence of wind disruption, emissions from "dirty pushes," i.e., heavy plumes, have a tendency to "roll out" from under the shed near the ends. The only two solutions to this problem are

1) to close the ends insofar as possible to interrupt wind currents, and 2) to extend the overall shed structure beyond the ends of the battery. Sheds can be designed to extend beyond the battery toward the quench tower to capture emissions during travel of the hot car.

8.1.2 Inspection

In the inspection of a coke-side shed, the inspector should consider the following:

1. The integrity of the shed structure and any missing roof panels or holes in the roof.
2. Leakage between the battery top and the shed interface.
3. Integrity and cleanliness of the baffle plates and heat shields.
4. Effect of cross-drafts and heavy-particle fallout.
5. The time required for the plume to clear under the shed.
6. Efficacy of capture and evacuation of emissions resulting from pushes from the ovens near the ends of the shed.
7. The greenness of the push; a shed that can contain clean pushes will not necessarily be able to contain the much larger gas volume generated during a dirty push.

8.2 ELECTRIC ARC FURNACE VENTILATION

8.2.1 Description

Canopy hoods are widely used to control secondary emissions from electric arc furnace (EAF) refining. The advantages of these hoods are that they do not interface with the furnace; they provide ventilation during charging, tapping, and slagging off; they do not affect furnace metallurgy; and their maintenance costs are low. The disadvantages are that they require high air volume; they are subject to cross-flow air currents that interface with fume control; and the plume can interfere with crane operator's line of vision during charging and tapping.

The need for high air volumes can be overcome by combining the canopy with a partial enclosure, as illustrated in Figure 40. The three-sided enclosure reduces the effect of cross-flow interference and directs the plume toward the hood. As shown earlier in Figure 6 and explained in the discussion regarding coke oven sheds, another approach is to use a baffled hood that concentrates suction towards the heaviest area of emissions.

The ultimate expansion of this approach is total furnace enclosure which was previously discussed in Section 7.

A complicating factor in EAF ventilation is the control system required to combine fourth-hole evacuation and a canopy hood on the same fan system. A balanced draft is difficult to achieve, and it must be established by trial and error. For example, the canopy hood is often dampered off when the furnace cover is in place so that suction will be concentrated on the fourth-hole takeoff. In this configuration, the control system directs total system suction to the canopy hood when the cover is off for charging and when the furnace is tilted for tapping. In some cases, however, the charging emission plume is still rising toward the hood while the cover is being quickly replaced. Thus, when the cover has been replaced and suction is taken off the canopy hood, the rising plume is not captured. In such an arrangement, it would be desirable either to replace the cover more slowly or to build an electrical delay into the automatic damper control system.

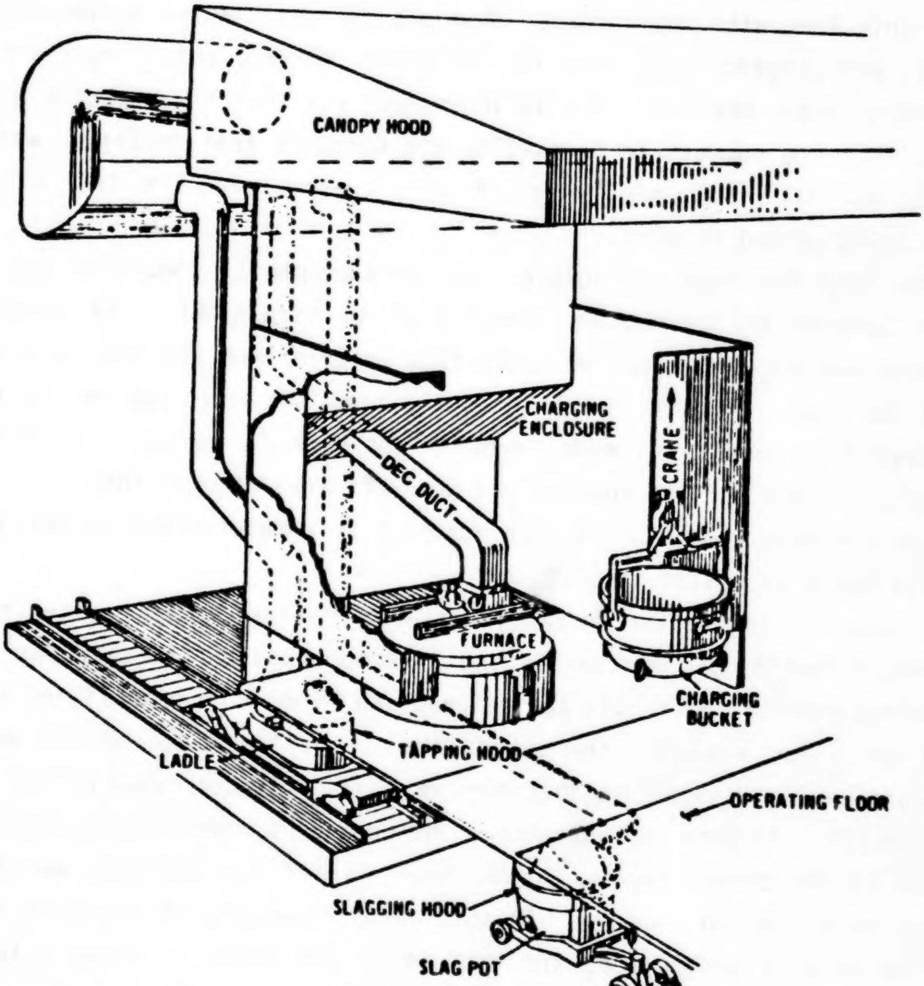

CANOPY HOOD
CHARGING ENCLOSURE
DEC DUCT
CRANE
FURNACE
CHARGING BUCKET
TAPPING HOOD
LADLE
OPERATING FLOOR
SLAGGING HOOD
SLAG POT

Figure 40. Electric arc furnace utilizing partial enclosure.

Source: U.S. Environmental Protection Agency, Office of Air Quality Planning and Standards, Research Triangle Park, North Carolina 27711

Figure 41 shows the basic types of ventilation systems used on electric arc furnaces. Figure 42 illustrates a popular approach in which the fourth-hole direct evacuation is combined with a canopy hood. In some cases, a third stage of capture is included by closing off the roof monitors in the area of the furnace and connecting a take-off duct to the roof. This third stage captures smoke that has escaped the canopy.

In the combined direct evacuation canopy system, baghouse temperature is controlled by mixing the cool gas from the canopy with the hot gas from the furnace. A cool-air emergency inlet damper is usually located prior to the baghouse. If this damper remains open, it will cause a ventilation "short-circuit" and rob the hood of suction.

As shown in Figure 43, two or more furnaces are typically combined on a system, which can produce complex flow balancing requirements. Table 13 presents an example of the ideal flow balance on a two-furnace system. Actually, flows through various duct sections must be monitored to confirm their desirability. All elements of the system must be kept in good condition to maintain the ideal balance. Misalignment of the gap in the DSE duct (see Figure 43) will result in the indraft of more combustion air at this point and lower suction elsewhere. Figure 44 summarizes total flow rate requirements for various EAF control approaches and indicates the rapid increase in flow requirements with the move from direct evacuation to building evacuation. These data are based on averages of various systems and should be used only as a general indicator of reasonable flow requirements.

Compromises were made in most plants where hoods had to be retrofitted. The furnace and bay configuration, clearance for canopy hoods above the crane gantry level, and ductwork layout became important factors because they could mitigate the efficiency of the canopy hood. Failure to consider activities that increase the fume generation rate, such as oxygen blowing and oxyfuel burners, during the initial design of the hoods can cause the system to be overloaded and capture efficiency to be reduced.

8.2.2 Inspection

The inspector should consider the following when inspecting EAF's:

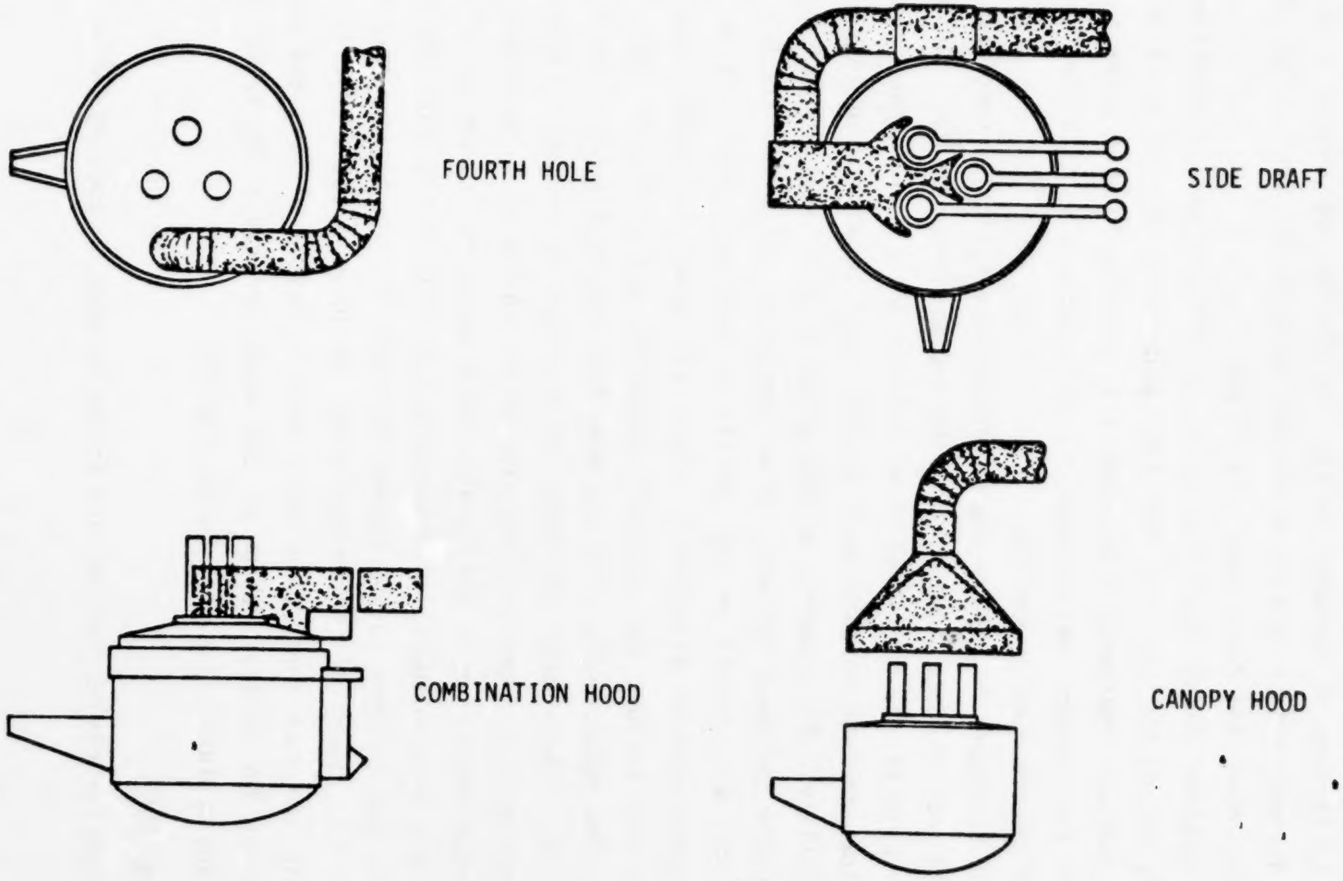

FOURTH HOLE
SIDE DRAFT
COMBINATION HOOD
CANOPY HOOD

Figure 41. Ventilation systems for electric arc furnaces.

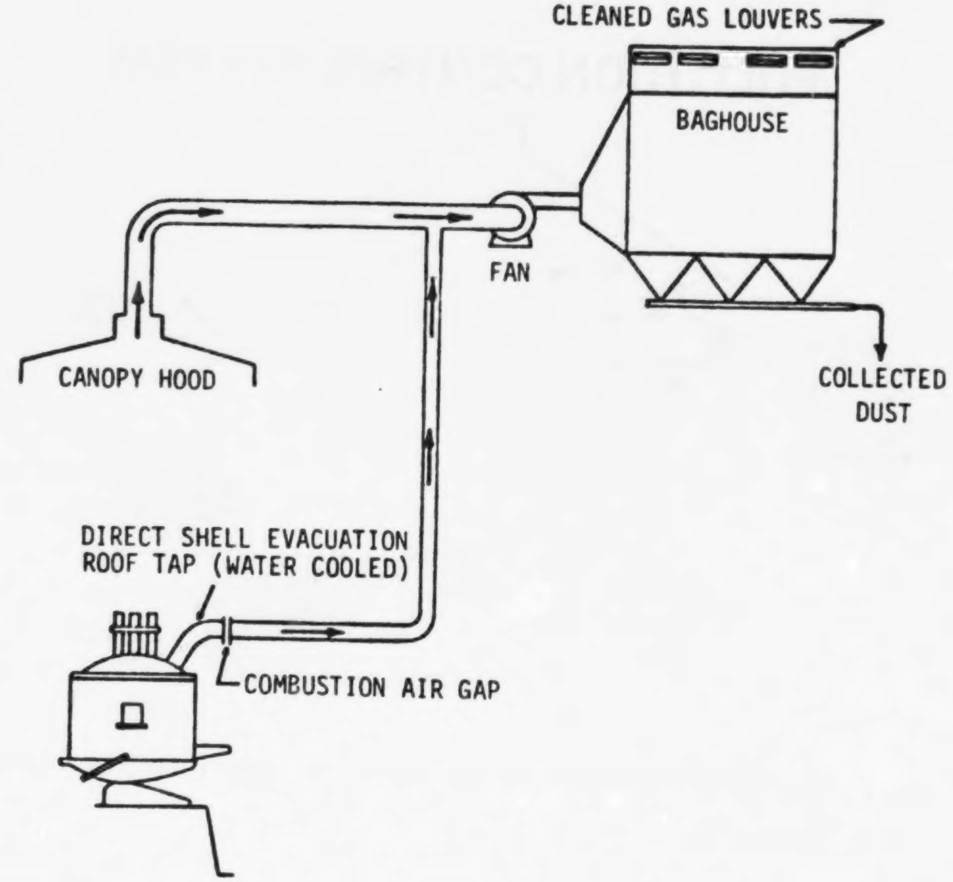

CLEANED GAS LOUVERS
BAGHOUSE
FAN
CANOPY HOOD
COLLECTED DUST
DIRECT SHELL EVACUATION ROOF TAP (WATER COOLED)
COMBUSTION AIR GAP

Figure 42. Combined direct shell evacuation with canopy hood.

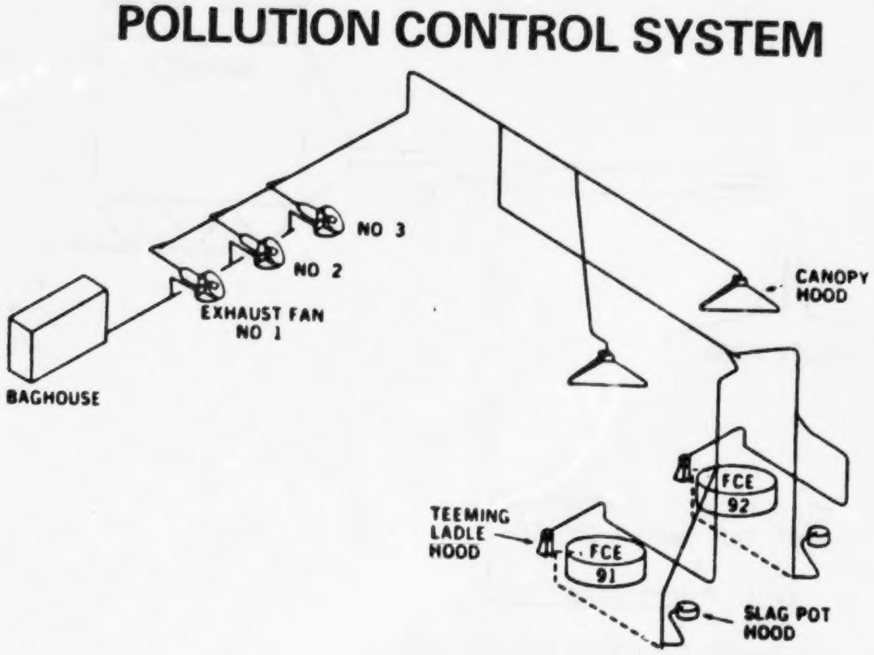

POLLUTION CONTROL SYSTEM
NO 3
NO 2
EXHAUST FAN NO 1
BAGHOUSE
CANOPY HOOD
TEEMING LADLE HOOD
FCE 92
FCE 91
SLAG POT HOOD

Figure 43. Multiple pickup points vented to common control device.

TABLE 13. EXAMPLE OF FLOW BALANCING OF MULTIPLE EVACUATION SYSTEM ON ELECTRIC ARC FURNACE

Item	Air flow, cfm
Furnace 1 charging	800,000
Furnace 1 slagging hood leakage	12,000
Furnace 1 tapping hood leakage	18,000
Furnace 1 direct shell leakage	25,000
Furnace 2 oxygen lancing	265,000
Furance 2 slagging (simultaneous with oxygen blowing)	118,000
Furnace 2 canopy leakage	44,000
Furnace 2 tapping hood leakage	18,000
Total system capacity	1,000,000

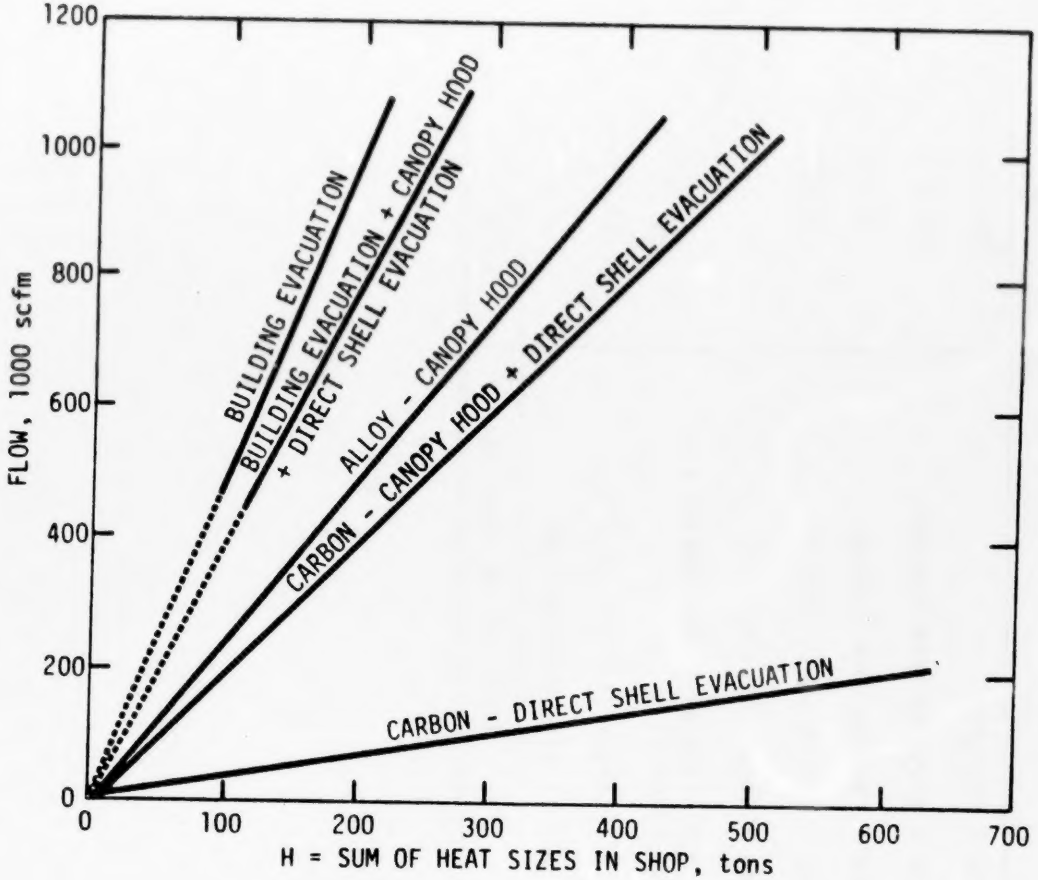


Figure 44. Flow rate required for electric arc furnace control.

1. The integrity of the canopy hoods, the side-draft hood, and all the duct work leading to the cleaning unit. In the case of an enclosed or sealed roof, any large holes or missing panels will reduce the capture efficiency.

2. Additional thermal currents produced by ancillary operations such as overflow of slag and/or metal on dumping ladles should be noted. These operations increase the load on the evacuation system.

3. Furnaces and bay layout have an important bearing on the capture efficiency of the hoods.

4. Multiple-furnace operations, additional pickup points, and the overall flow balance under different modes of operation should be observed.

5. Crosswinds inside the shop can severely disrupt the capture efficiency of the hoods.

8.3 BLAST FURNACE CASTHOUSE CONTROL

8.3.1 Description

Although recently developed technology[24] does not use ventilation for control of casting emissions, numerous ventilation-based systems are installed or planned on blast furnace casthouses. Sources of fugitive emissions in the casthouse are the tap hole, trough, skimmer, runners, hot metal spouts, and slag spouts. These various sources require a multiple hooding system or total building evacuation. Local control is favored because total flow requirements are less; however, the design of many older casthouses precludes their use. Table 14 lists the typical flow volume requirements for a large blast furnace casthouse system.

These systems require flow balancing, proper design, and the other factors discussed earlier. Also, the frequent removal and replacement of close-fitting hoods or runner covers invite both damage and misplacement. Because of the proximity of the hooding to the molten metal, frequent maintenance of the refractory linings and runner covers is required. Large blast furnaces with multiple tapholes and casthouses require a carefully designed flow balancing and damper system for switching from one casthouse to the other. Such systems have been designed for full evacuation of one taphole at a time. If damper settings are such that two tapholes are evacuated simultaneously, capture efficiency will be curtailed.

TABLE 14. BLAST FURNACE CASTHOUSE TYPICAL VOLUME REQUIREMENTS

Location	Volume requirements, cfm	Volume requirements, m^3/min
Taphole	70,000	1978
Skimmer	35,000	989
Iron runner	70,000	1978
Hot metal spouts	75,000	2119
Slag spouts	50,000	1413
Total	300,000	8477

8.3.2 Inspection

When inspecting blast furnace casthouse systems, the inspector should consider the following:

1. The performance of the taphole hood during various stages of the tap (i.e., immediately after taphole opening, during heavy metal flow, and during slag and metal flow). The capture efficiency of the hoods (covers) at the trough, skimmer, runners, and slag and metal spouts also should be observed during various stages of the cast and under different conditions, such as high and low metal temperature and high and low slag basicity.
2. When there are multiple casthouses, the flow balance between the casthouses should be observed. The flow is sometimes equally split between the casthouses, which significantly reduces the capture efficiency.
3. The furnace layout, casthouse layout, and ductwork location have an important bearing on the efficiency of the evacuation system.
4. Cross-currents inside the casthouse mitigate the hood capture. It is common practice to remove and replace the siding in the casthouse, depending on the season. A system designed for a closed structure is not likely to perform adequately if the structure is opened.

8.4 CONTROL SYSTEMS ON BASIC OXYGEN FURNACES (BOF's)

8.4.1 Description

Primary capture systems on BOF's use a specially designed hood over the mouth of the furnace. This usually operates effectively when the furnace is in an upright position; however, when the furnace is tilted for charging, tapping, or testing, emissions generally escape the hood (for the reasons discussed earlier). Capture of these secondary emissions requires additional hooding or enclosures. The total enclosure of BOF was previously discussed in Section 7.

Water-cooled primary hoods are sometimes used to generate steam because the gas temperatures in the hood are initially over 3000°F. These hoods are exposed to the most severe operating conditions of all the process applications discussed--fluctuating high temperatures and the possibility of physical damage from cranes or ejections of material from the furnace. The hood may be elevated several feet above the furnace to allow indraft of combustion air (open), or they may be closely mated with the furnace to conserve the carbon monoxide inherent in the exhaust gas (closed). Figure 45 illustrates these BOF hood arrangements.

Many different hood constructions have been tried to cope with the severe operating conditions. The membrane or "tube-bar" type is usually favored today. Table 15 presents the performance characteristics of various hood constructions. Table 16 lists the various hood types that are used for full combustion and limited combustion systems in the United States, Canada, and Mexico.[25]

Control of secondary emissions is achieved by local hoods or enclosures. Local hoods are generally ineffective because they are too far from the source, are too small to encompass the emissions, and because the evacuation rate is usually insufficient. Figure 46 illustrates a simple external hooding arrangement for charging emissions.

Figure 47 illustrates the Gaw damper approach, which enables the primary hood to function as a secondary hood for charging emissions. The movable damper increases hood suction by restricting the hood face area. Maintenance of the damper is high because of the harsh environment. As in the case of the local hood discussed previously, effective use of the Gaw damper requires a slow pouring rate of charged metal to decrease fume generation and a minimum tilt angle of the vessel to generally direct the fumes toward the hood.

8.4.2 Inspection

When inspecting BOF ventilation systems, the inspector should consider the following:

1. Primary hood capture efficiency during operations such as slopping.
2. The tilt angle and hot metal pour rate during hot metal charging, which greatly affects the capture efficiency.
3. The performance and capture of secondary controls.
4. Emissions from the lance hole or fume rollout (puffing) from the primary hood, which indicate insufficient draft on the hood.
5. In closed hood systems, the snug fit of the hood skirt at the furnace. (Indraft of air above the design level will overload the fan and cause the variable-throat scrubber to open, which reduces pressure drop.)

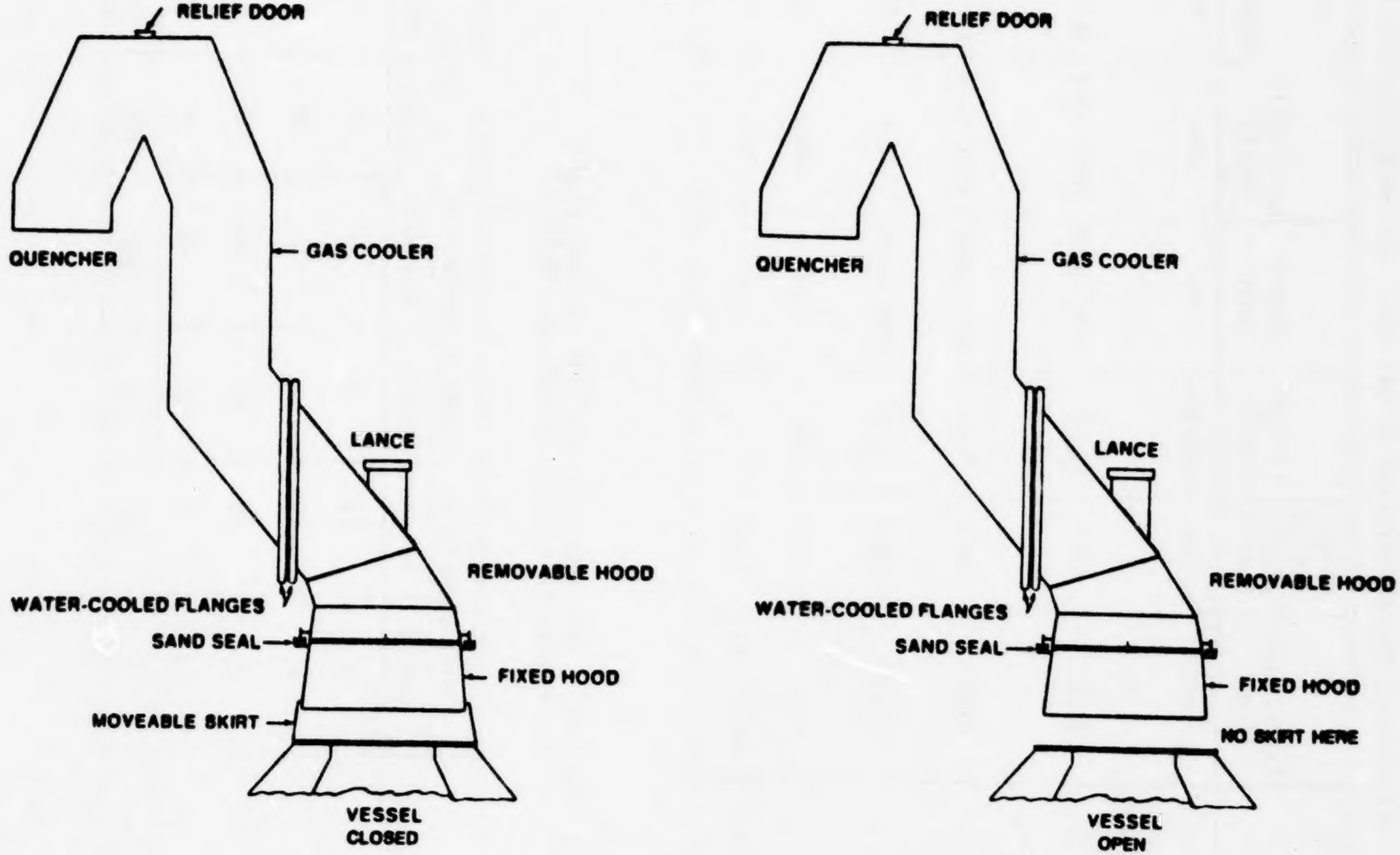

RELIEF DOOR
QUENCHER
GAS COOLER
LANCE
REMOVABLE HOOD
WATER-COOLED FLANGES
SAND SEAL
FIXED HOOD
MOVEABLE SKIRT
VESSEL CLOSED
RELIEF DOOR
QUENCHER
GAS COOLER
LANCE
REMOVABLE HOOD
WATER-COOLED FLANGES
SAND SEAL
FIXED HOOD
NO SKIRT HERE
VESSEL OPEN

Figure 45. BOF hood arrangements.

Courtesy: BOF Steelmaking, Volume III. Published by American Institute of Mining, Metallurgical and Petroleum Engineers.

TABLE 15. PERFORMANCE CHARACTERISTICS OF DIFFERENT BOF HOOD CONSTRUCTIONS[25]

	Refractory-lined	Water-cooled plate panels	Formed panels	Double pass	Water wall boiler	Membrane
Initial cost	Lowest	Low	Moderate	High	High	High
Ability to take high temperature	Poor	Fair	Good	Very good	Very good	Very good
Ability to take temperature change	Poor	Fair	Good	Very good	Very good	Very good
Resistance to slag buildup	Poor	Good	Good	Very good	Fair	Good
Resistance to scaling	--	Poor	Fair	Good	Good	Good
Maintenance Cost	Very high	High	Fair	Low	Fair	Fair

TABLE 16. BOF HOOD CONSTRUCTION DESIGNS IN USE IN THE UNITED STATES, CANADA, AND MEXICO[25]

Construction	Full combustion		Limited combustion		Total	
	Number	Percent	Number	Percent	Number	Percent
Tube	10	12	3	15	13	12
Panel/jacket	41	48	4	20	45	43
Membrane	30	35	13	65	43	41
Not known	4	5	0	0	4	4
Total	85	100	20	100	105	100

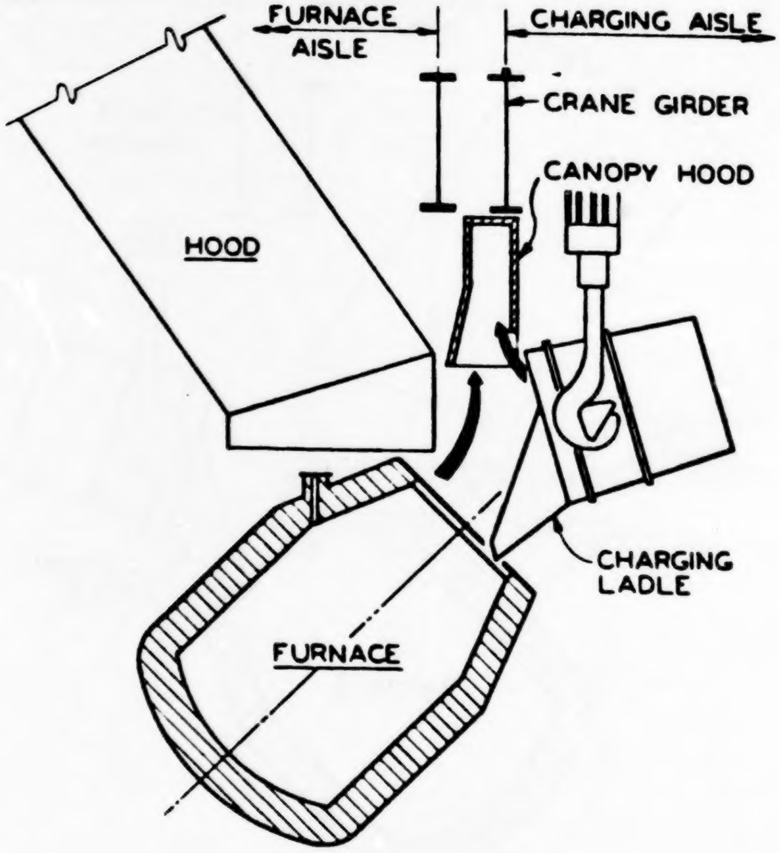

FURNACE AISLE
CHARGING AISLE
CRANE GIRDER
CANOPY HOOD
HOOD
CHARGING LADLE
FURNACE

Figure 46. Canopy hood concept for BOF charging emissions.

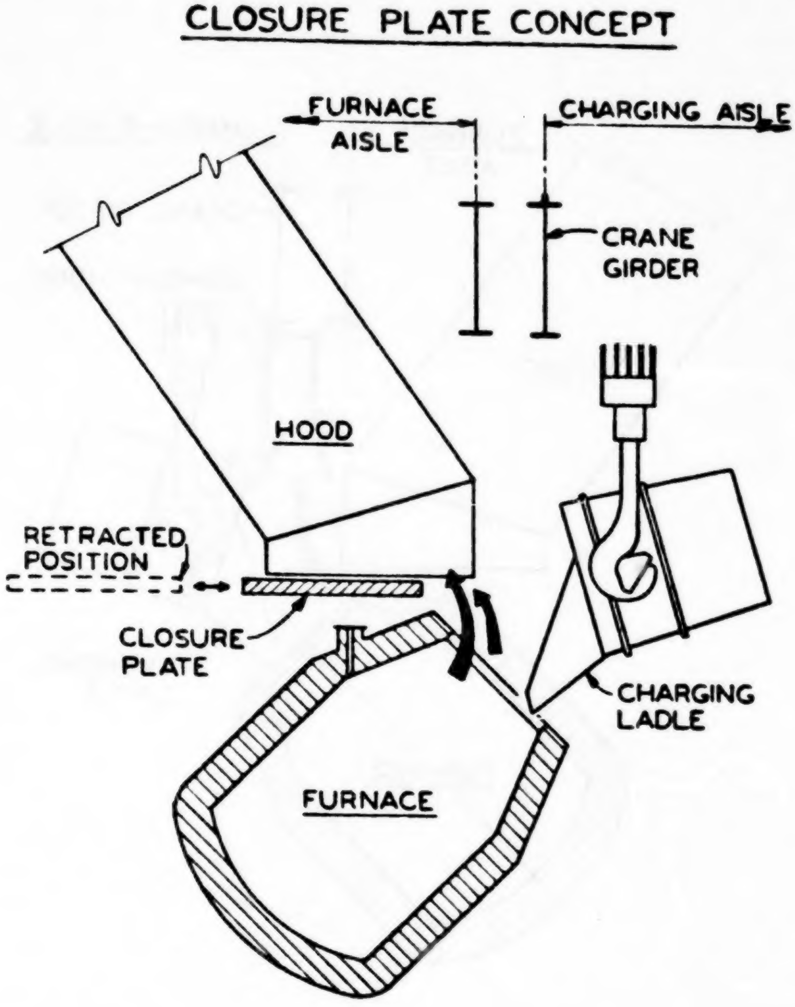

CLOSURE PLATE CONCEPT
FURNACE AISLE
CHARGING AISLE
CRANE GIRDER
HOOD
RETRACTED POSITION
CLOSURE PLATE
CHARGING LADLE
FURNACE

Figure 47. Gaw damper (closure plate) use in BOF control.

8.5 BUILDING EVACUATION

The use of total building evacuation to control air pollution contamination or to improve the work area environment is normally a difficult and costly approach.

The problem areas are:

1. Achieving the necessary number of air changes per hour (20 or more, depending on the contaminants).
2. The ability to reach the specific work areas to supply these air changes. (Some areas have dead air pockets, and either ducting or forced ventilation has to be provided.)
3. The volume of air required. (A building 800 ft long, 80 ft wide, and 40 ft high has a volume of 2,560,000 ft^3. With 20 air changes per hour, ≃850,000 scfm of air is required. If contaminants, open doorways, heat emissions, and air currents caused by thermal processes or vehicle movement are added, the actual requirement for effective ventilation of such a building would be 3 to 4 x 10^6 scfm.)
4. The existing building structure not being designed for total enclosure. (The weight of siding, ducting, and wind loads may require strengthening of the building columns and roof trusses and the addition of more purlins, struts, and bracing.)
5. The location of process equipment and the flow patterns of materials not being conducive to the proper collection of emissions. (Lighting may also be a problem.)
6. Need for redundancy of the fans. (Clean-out mechanisms should be built into the ducting that are readily accessible and repairable. Ducting is long and huge.)

A newly designed building in which process equipment is located specifically for total building evacuation can greatly reduce these problems.

A variation of building evacuation that does not involve local hooding and ventilation systems is the roof-mounted electrostatic precipitator (REP).[26] This system has not yet been applied in the United States, but it has been successfully used in Japan.[26] The technique involves roof modification to help channel the natural plume rise of BOF fugitive emissions into an REP, which collects the process fugitive emissions. The system has no fan, and the plates are generally cleaned by use of a water spray.[26] The REP specification provided by Sumitomo Heavy Industries for controlling two BOF's, each with 300-ton capacity, offers a design efficiency of 91.5 percent.[26]

8.6 COPPER CONVERTERS

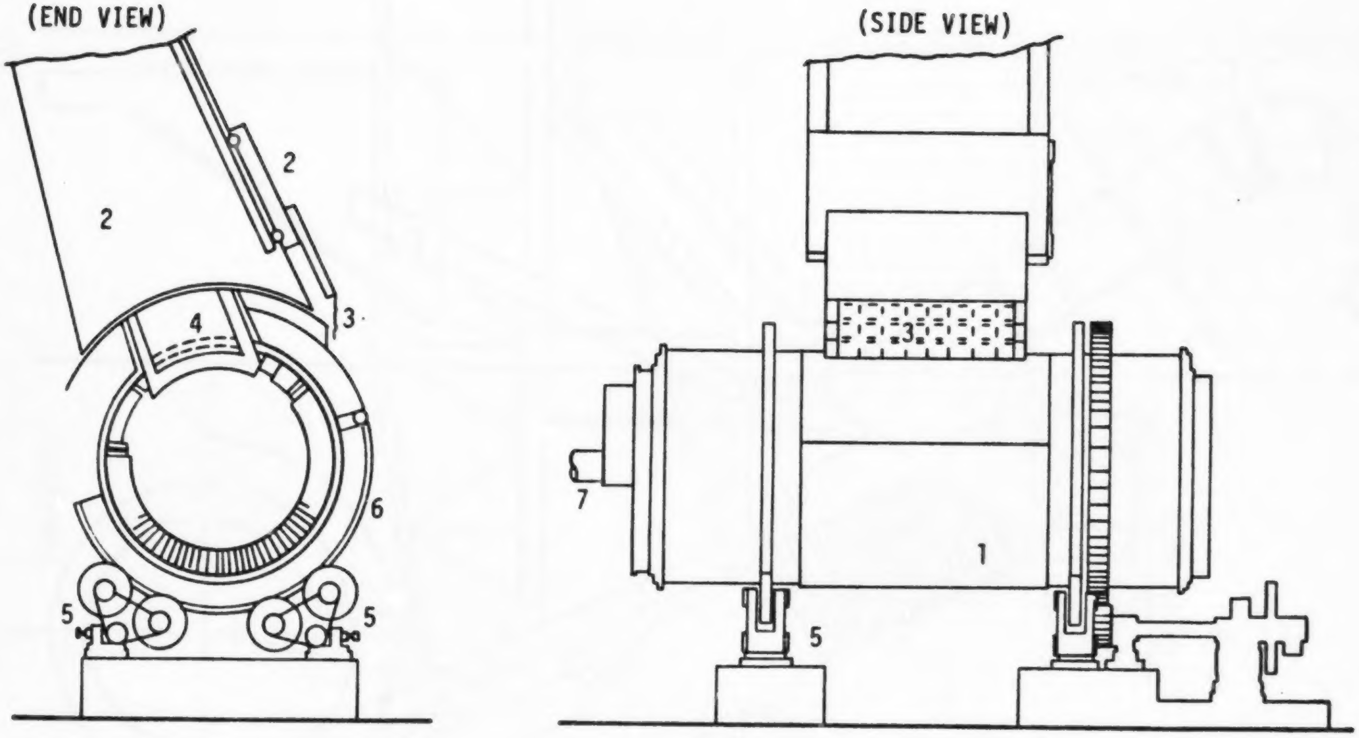
8.6.1 Description

Capture of emissions from Peirce-Smith type copper converters presents some unique problems. The converter is essentially a horizontal cylinder with a circular opening on top. This cylinder is rotated around its longitudinal axis so the opening can face toward the ladle. This rotation makes it impossible to achieve an absolutely tight seal between the hood and the converter. Also, when the converter is rotated for charging and tapping, the opening is no longer under the hood, and fumes tend to escape into the converter building. The heat and collisions with the ladle suspended from the crane tend to cause the hood to warp, and eventually it does not fit well. Figure 48 shows a converter and the primary hood.

At copper plants, the emissions from all converters (usually three to five units) are ducted together to a common particulate control device and sometimes to a sulfuric acid plant. Dampers in the individual breeching control the draft on any single converter. These dampers require routine maintenance to ensure proper functioning.

Primary converter hoods have sliding gates on the front side as shown in Figure 48. These gates are really movable hood extensions that serve to cover the converter opening and improve capture efficiency.

Some smelters have installed secondary hoods (Figure 49) or air curtain systems to capture fumes when rotation causes the converter opening not to be under the main hood. By providing a strong blast of air to blow fumes into a suction hood, the air curtain provides an open area for locating the charging and tapping ladle and the crane hooks and cables. Secondary hoods suspended above the converter opening and ladle have the same weaknesses found in other processes. For example, because of their distance above the converter and ladle, they are affected by thermal and cross-drafts and are not effective unless face velocities of the hood are high.


(END VIEW)
(SIDE VIEW)
1
2
2
2
3
3
4
5
5
5
6
7

(1) SHELL; (2) HOOD; (3) AIR BAFFLE; (4) CONVERTER MOUTH; (5) ROLLERS;
(6) TURNING RINGS; (7) AIR - SUPPLY DUCT;

Reprinted from the Kirk-Othmer Encyclopedia of Chemical Technology, 3rd edition, Fig. 9, Page 836. Copyright © John Wiley & Sons, Inc. 1979.

Figure 48. Peirce-Smith converter.

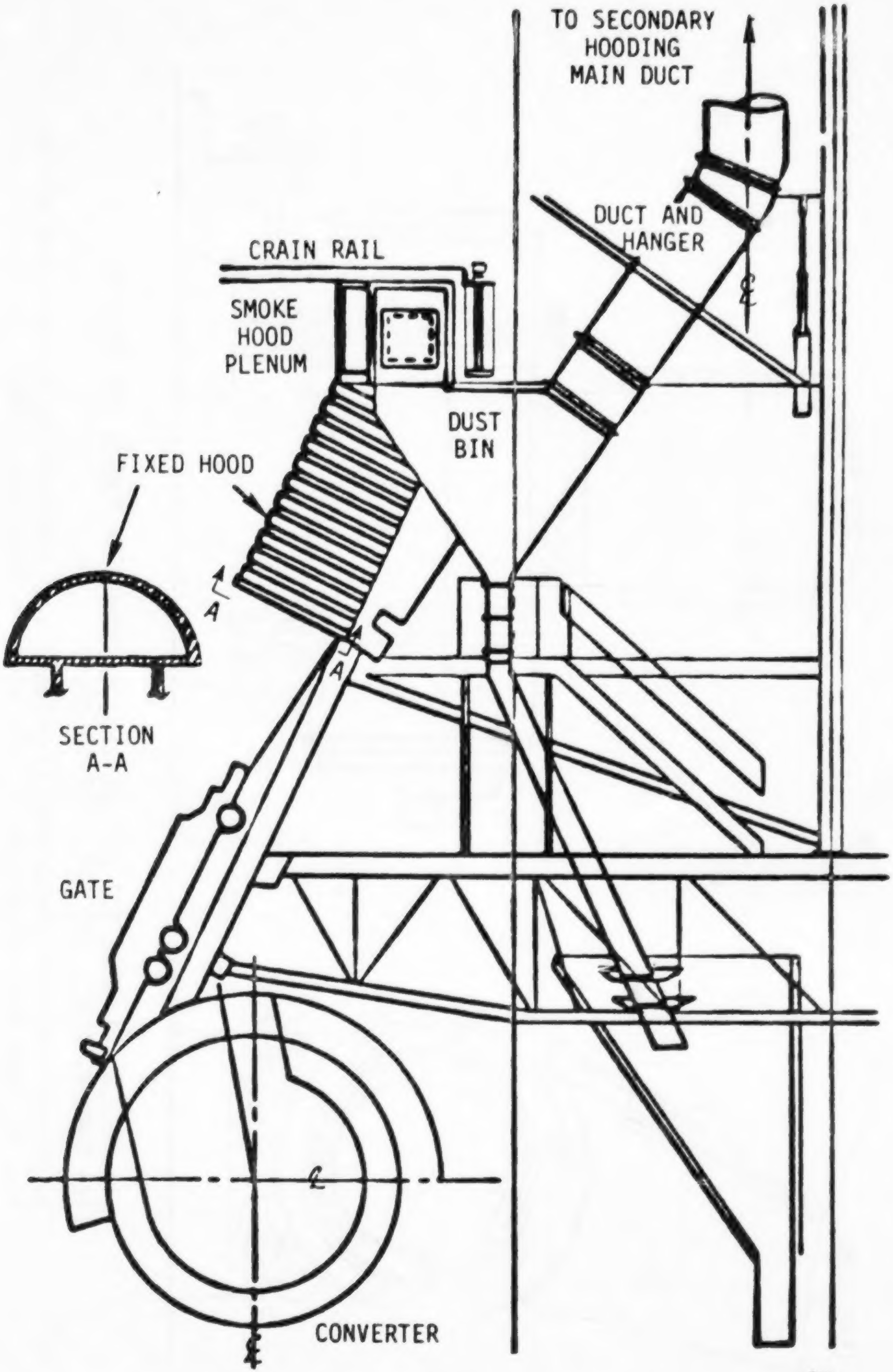

TO SECONDARY
HOODING
MAIN DUCT
DUCT AND
HANGER
℄
CRAIN RAIL
SMOKE
HOOD
PLENUM
DUST
BIN
FIXED HOOD
A
A
A
SECTION
A-A
GATE
℄
CONVERTER
℄

Figure 49. Secondary converter hood configuration.[27]

8.6.2 Inspection

Converter hoods are inspected by making a visual assessment of their physical condition and the emissions that escape during the various converter cycles. The hood and duct system should not contain any openings that allow air to be drawn in and reduce the suction at the converter. The hood lip should come within about 6 inches of the converter opening during blowing. The hood gate or slide should be extended when the converter is in the upright position to ensure that the converter opening is covered. During converter rollout, the draft should be reduced so that a high-suction air flow can be maintained on the converters in the blowing mode.

REFERENCES

1. American Conference of Governmental Industrial Hygienists. Industrial Ventilation. 16th Ed. 1980.

2. Hemeon, W. C. L. Plant and Process Ventilation. The Industrial Press, New York. 1963.

3. Department of Health, Education and Welfare. Air Pollution Engineering Manual. J. A. Danielson (Ed.) Public Health Service Publication 999-AP-40, 1967.

4. Buffalo Forge Company. Fan Engineering. 8th Ed. Buffalo, New York. 1983.

5. Jablin, R., and D. W. Coy. Engineering Study of Roof-Mounted Electrostatic Precipitator for Fugitive Emission Control on Two Basic Oxygen Furnaces of 300-ton capacity. In: Proceedings of Symposium on Iron and Steel Pollution Abatement Technology for 1981. EPA 600/9-82-021, December, 1982. pp. 120-138.

6. Coy, D. W., and L. E. Sparks. Roof-Mounted Electrostatic Precipitators - Visible Emissions Evaluation and Computer Modeling Performance Predictions. Presented at the 1983 Symposium on Iron & Steel Pollution Abatement Technology.

7. Vajda, S. Blast Furnace Casthouse Emission Control Without Evacuation. In: Proceedings of Symposium on Iron and Steel Pollution Abatement Technology. EPA 600/9-83-016, November 1982. pp. 232-240.

8. Dixon, T. E., and H. Nomine. Capital and Operating Costs of OBM/Q-BOP Gas Cleaning Systems. Iron and Steel Engineer, March 1978. p. 37.

9. U.S. Department of Health Education and Welfare. Air Pollution Engineering Manual. J. A. Danielson (Ed.). Public Health Service Publication 999-AP-40, 1967. Chapter 3.

10. Dalla Valle, J. M. Exhaust Hoods. The Industrial Press, New York. 1946.

11. Kriechelt, T. E. et al. Natural Ventilation in Hot Process Buildings. Iron and Steel Engineer, December 1976.

12. Hemeon, W. C. L. Plant and Process Ventilation. The Industrial Press, New York. 1955.

13. Davis, J. A. Unidirectional Flow Ventilation System. Presented at the 140th Annual AIME Meeting, New York, February 18, 1975.

14. American Conference of Governmental Industrial Hygienists. Industrial Ventilation, A Manual of Recommended Practice. 17th Ed. 1982.

15. U.S. Environmental Protection Agency. Air Pollution Engineering Manual. 2nd Ed. AP40, May 1973.

16. Pollak, R. Selecting Fans and Blowers. Chemical Engineering, January 22, 1973.

17. Fan Engineering. 8th Ed. Published by Buffalo Forge Company, New York. 1983.

18. Nicola, A. G. Best Available Control Technology (BACT) for Electric Furnace Shop Emission Control. 36th Electric Furnace Conference Proceedings, Vol. 36, Toronto December 1978, Sponsored by The Electric Furnace Division of I&S Society of American Institute of Mining, Metallurgical and Petroleum Engineers.

19. Brand, P. G. A. Current Trends in Electric Furnace Emission Control. Iron and Steel Engineer, February 1981. p. 59.

20. Henninger, J. L. et. al. Closing In On Arc Furnace Emissions at Carpenter Technology. Iron & Steel Engineer. March 1984.

21. Canefield, G. Jr. Pollution Control Using a Furnace Enclosure. 36th Electric Furnace Conference Proceedings, Vol. 36, Toronto December 1978, Sponsored by The Electric Furnace Division of I&S Society of American Institute of Mining, Metallurgical and Petroleum Engineers.

22. U.S. Environmental Protection Agency. Revised Standards for Basic Oxygen Process Furnaces - Background Information for Proposed Standards. Preliminary Draft. 1980.

23. Roe, et al. United States Patent No. 3,844,901. October 29, 1974.

24. Vajda, S. Blast Furnace Casthouse Emission Control Without Evacuation. Iron and Steel Engineer, June 1983. p. 29.

25. American Institute of Mining, Metallurgical and Petroleum Engineers. BOF Steelmaking. Vol. III. 345 E. 47th Street, New York.

26. Jablin, R., and D. W. Coy. Engineering Study of Roof-Mounted Electrostatic Precipitator (REP) for Fugitive Emission Control on Two Basic Oxygen Furnaces of 300-ton Capacity. In: Proceedings of Symposium on Iron and Steel Pollution Abatement Technology for 1981. EPA 600/9-82-021, December 1982. pp. 120-138.

27. U.S. Environmental Protection Agency. Control of Copper Smelter Fugitive Emissions. EPA-600/2-80-079, May 1980.

APPENDIX A

BIBLIOGRAPHY

Adams, R. W. 1976. Waste Gas Handling. In: BOF Steelmaking, Volume 3, Design. Iron and Steel Society of AIME.

American Iron and Steel Institute. 1965. Steel Mill Ventilation.

Ban, E. 1980. Innovations and Improvements in the Ore Sintering Process for Air Pollution Control. In: Proceedings of the First Symposium on Iron and Steel Pollution Abatement Technology, Chicago, Illinois, October 30-November 1, 1979. EPA-600/9-80-012.

Baturin, V. V. 1972. Fundamentals of Industrial Ventilation. 3rd Edition, Pergamon Press.

Bender, M. 1979. Fume Hoods, Open Canopy Type-Their Ability to Capture Pollutants in Various Environments. American Industrial Hygiene Association Journal. pp 119-127.

Bender, M., and W. D. Baines. 1976. Operation of an Open Canopy Fume Hood in a Crossflow. ASME Ser., 97(2):242-243.

Blast Furnace Cast House Emission Control Technology Development in the USSR. Translated by SCITRAN for U.S. Environmental Protection Agency.

Blessing, K. E. 1963. Electric Furnace Fume Control. Chemical Engineering Progress. 59:60-64.

Brand, P. G. A. 1981. Current Trends in Electric Furnace Emission Control. Iron and Steel Engineer, 58:59.

Brough, J. R. 1964. Air Pollution Control of An Electric Furnace Steelmaking Shop. Journal of the Air Pollution Control Assn. 22:170.

Brown, C. M. 1980. Control of Secondary Emissions from Basic Oxygen Steelmaking Furnaces. Iron and Steel Engineer. 57:39.

Buffalo Forge Co. 1970. Fan Engineering.

Burroughs, J. D. et al. 1976. Application of Enclosed Coke Pushing Emission Controls at Granite City Steel. In: Proceedings of 69th Annual Meeting of Air Pollution Control Association, Portland, Oregon, June 27-July 1, 1976.
Calderon, A. 1977. Controlling Emissions from Coke Ovens. Iron and Steel Engineer, 54(8):42.

Campbell, W. W. 1962. Development of an Electric Furnace Dust-Control System. Journal of the Air Pollution Control Assn. 12:576.

Canfield, G., Jr. 1978. Pollution Control Using a Furnace Enclosure. In: Proceedings of the Electric Arc Furnace Conference, Vol. 36. Toronto Meeting. December 5-8, 1978.

Cheever, C. L., et al. 1972. Proceedings (two volumes) of 12th Air Cleaning Conference, Oak Ridge, Tennessee.

Cheremisinoff, P. N. and N. P. 1976. Calculating Air Flow Requirements for Fume Exhaust Hoods. Plant Engineering, 30(4):111-114.

Cheremisinoff, P. N. and N. P. 1976. Calculating Air Volume Requirements for Fume Exhaust Hoods--Nonenclosure Types. Plant Engineering, 30(6):143-144.

Clarke, J. H. Practical Industrial Ventilation Design. A collection of article reprints, Heating, Piping, and Air Conditioning Journal.

Coleman, R. T., Jr., and R. Vandervort. 1980. Demonstration of Fugitive Emission Controls at a Secondary Lead Smelter. In: Proceedings of World Symposium of Metallurgy and Environmental Control, Las Vegas, February 24-28, 1980.

Conners, A. 1980. Developments in Coke-Oven Emission Control. Iron & Steel Engineer, 57(6):33.

Coy, D. W., and R. Jablin. 1980. Review of Foreign Air Pollution Control Technology for BOF Fugitive Emissions. In: Proceedings of the First Symposium on Iron and Steel Pollution Abatement Technology, Chicago, Illinois, October 30-November 1, 1979. EPA-600/9-80-012.

Current, P. 1980. Sinter Plant Windbox Gas Recirculation and Gravel Bed Filtration Demonstration. In: Proceedings of the First Symposium on Iron and Steel Pollution Abatement Technology, Chicago, Illinois, October 30-November 1, 1979. EPA-600/9-80-012.

Drivas, P. J., et al. 1972. Experimental Characterization of Ventilation Systems in Buildings. Environmental Science Technology, 6(7):609-614.

Eisenbarth, M. J. 1981. Fume Extraction Hoods in the Iron & Steel Industry. In: A Specialty Conference on Air Pollution Control in the Iron and Steel Industry, April 21-23, 1981.

Emswiler, J. E. 1926. Neutral Zone in Ventilation. ASHRAE Transactions, Vol. 32.

Escudier, and J. Maxworthy. 1973. On the Motion of Turbulent Thermals. Fluid Mechanics, V. 61 - Pt. 3. pp 541-552.

Field, A. A. 1977. New Concepts in Industrial Air Replacement. Heating, Piping, and Air Conditioning, 49(3):84, 86, 88, 90.

Fields, S. F., et al. 1982. Modeling of Hood Control of Blast Furnace Casting Emissions. In: Proceedings of the Symposium on Iron and Steel Pollution Abatement Technology for 1981, Chicago, Illinois, 10/6-10/8/81. EPA-600/9-82-021.

Flannigan, L. J., et al. 1974. Development of Design Criteria for Exhaust Systems for Open Surface Tanks. Prepared for NIOSH, Cincinnati, Ohio. NIOSH-75/108.

Flux, J. H. 1974. Containment of Melting Shop Roof Emissions in Electric Arc Furnace Practice. Ironmaking and Steelmaking (Quarterly). No. 3, pp 121-133.

Forestier, G., and X. Tinchant. 1981. Cooling of Oxygen Converter Gas, Efficiency of the Exhaust Hood, and the Stack of the Gas collection Equipment. Tech. Rev. Metall., 78(12):937-948.

Franza, M. E. 1982. Controlling Fugitive VOC Emissions From the Metal Finishing Industry. Met. Finish, 80(12):39-45.

Fuller, H. 1981. The Alliance/Mitsubishi Pushing Emission Control System at Ironton Coke Corp. Iron and Steel Engineer, 58(5):29.

Goldman, J. Technical Support Document for Control of Coke Oven Pushing Emissions. Research Triangle Institute. EPA Contract 68-01-4141, Task 33.

Goodfellow, H. D. 1978. Fume System Design for Sidbec's Pellet Melting Furnaces. In: Proceedings of the Electric Arc Furnace Conference, Vol. 36. Toronto Meeting, December 5-8, 1978.

Great Lakes Carbon Shuns Naysayers, Uses Hood to Collect Smoke from St. Louis Coke Battery. 33 Magazine.

Haddrill, D. M. 1979. Control of Welding Fumes in Fabrication Shops. Metal Construction and British Welding J., 11(2):84-89.

Hanks, D. J. 1979. M/E Update: Fans and Blowers. Specif. Eng., 41(1): 129-134.

Hartung, Kuhn and Co. Pushing Emission Control System; Minister Stein. Dusseldorf, West Germany.

Hayashi, T., et al. 1976. Land Based, Mobile Haroley "Trar-L-Vent" Exhaust System for Coke-Side Pushing Emission Control. Hawley Manufacturing Corp. Indianapolis, Indiana. On the Elimination of Welding Fumes by Richard Hodd. Trans. Japan Weld. Soc., 7(1):18-26.

Helfritch, D. 1978. Arc Furnace Fume Control by an Electrostatic Fabric Filter. In: Proceedings of the Electric Arc Furnace Conference, Vol. 36. Toronto Meeting. December 5-8, 1978.

Hemeon, W. C. L. 1963. Plant and Process Ventilation, The Industrial Press, New York, New York, 2nd Edition.

Hoult, D. P. 1972. Turbulent Plume in a Laminar Cross Flow. Atmospheric Environment, Vol. 6, pp 513-531.

Iron and Steel Engineer. 1978. 55(3):83. Hooded Quench Car System Controls Coke Pushing Emissions.

Iron and Steel Engineer. 1980. American Air Filter. Local Capture System Controls. Coke Pushing Emissions. 57:69.

Irvine, H. B. 1980. Waste Heat Recovery From a Dust Collection System. In: Proceedings of 38th Electrical Furnace Conference, Pittsburgh, December 9-12, 1980.

Jacobs, C. O. 1979. Exhaust System for Electric Arc Welding Instruction. Presented at the Winter Meeting of ASAE, New Orleans, December 11-14, 1979.

Kaercher, L. T. 1974. Air Pollution Control for an Electric Furnace Melt Shop. Iron and Steel Engineer Year Book. pp 216.

Karnaukh, N. N., et al. 1973. Trapping of Gases From Ingot Molds in Pouring of Steel With Exothermic Mixtures. Metallugist (USSR), 17(3-4):189-190.

Kosarev, L. V., et al. 1978. Complex Systems of Technological Ventilation Apparatus for Collecting and Purifying Gases in Arc Furnaces. Metallurg., 21(5-6):26-29.

Kreichelt, T. W., and T. G. Keller. 1972. Roof Monitor Emissions; Test Methodology. Journal APCA, Vol. 22, No. 8.

Kreichelt, T. E., G. R. Kern, and F. G. Higgins, Jr. 1976. Natural Ventilation in Hot Process Buildings in the Steel Industry. Iron and Steel Engineer.

Krueger, D. 1964. Gas Cleaning in Iron and Steel Works. Blast Furnace and Steel Plant. 52:503.

Lawrie, W. G. 1970. Dust Collection in Industry. Filtr. Separ., 7(6):695-699.

Lusty, V. 1980. Collecting Emissions From Electric Induction Furnaces. Foundry Trade Journal, 148(3185):550, 553, 558, 562.

Malchaire, J. B. 1981. Design of Industrial Exhaust Systems Using Programmable Calculator. Annuals Occup. Hyg., 24(2):217-224.

Marchisio, I. C. 1981. Fiat/Teksid Goes "Snuff Box." Iron and Steelmaker. 8(6):36+.

Marchland, D. 1976. Possible Improvements to Dust Collection in Electric Steelplants, etc. Ironmaking and Steelmaking. No. 4, pp 221-229.

Marchland, D. 1981. Waste Gas Collection on Electric Arc Furnaces: BFI Research & Development During the Last 20 Years. In: A Speciality Conference on Air Pollution Control in the Iron and Steel Industry, April 21-23, 1981.

Martin J. R., A. D. Robertson, and A. T. Sheridan. 1974. Modeling Studies of Roof Extraction Systems for Arc Furnace Melting Shops. Presented at Conference: Quality of the Environment and the Iron and Steel Industry. Luxembourg, September 24-26, 1974.

Masany, T. J. 1982. Blast Furnace Casthouse Control Technology - Fall 1982 Update. In: Proceedings of the Symposium on Iron and Steel Pollution Abatement Technology for 1981, Chicago, Illinois, 10/6-10/8/81. EPA-600/9-82-021.

McCluskey, E. J. 1976. Design Engineering of the OH Gas Cleaning System at Inland's No. 2 BOF Shop. Iron and Steel Engineer, 53(12):53.

McCrillis, R. C. 1978. Basic Iron Blast Furnace Casthouse Emissions and Emission control. Paper presented at the 71st annual meeting of the Air Pollution Control Association at Houston, June 25-30, 1978.

McDermoth, H. J. 1976. Handbook of Ventilation for Contaminant Control. Ann Arbor Science Publishers, Ann Arbor, Michigan.

Mertz, J. W. and R. R. Pfahler. 1975. How to Troubleshoot Large Industrial Fans. Hydrocarbon Process, 54(6):57-62.

Molnar, J. 1978. Graphical Method for Determining Air Requirements of Exhaust Hoods. Plant Engineering, 32(4):132-133.

Mortimer, V. D., Jr., et al. 1982. In-Depth Survey Report of Valley Chrome Platers. Bay City, Michigan, April 27-May 1, 1981. Prepared for National Institute for Occupational Safety and Health, Cincinnati, Ohio.

Muermann, H. 1978. Ventilation and Air Conditioning of Modern Industrial Buildings. Wasser, Luft Und Betrieb, 22(4):126-128.

Natalizio, A., and C. Twiffe-Molecey. Ventilation of Mill Buildings - New Directions. Iron and Steel Engineer. 57(7):51-56.

Nelson, F. D., et al. 1976. Plant Layout and Material Handling. In: BOF Steelmaking, Volume 3, Design. Iron and Steel Society of AIME.

Nicola, A. G. 1979. Blast Furnace Cast House Emission Control. Iron and Steel Engineer. 56:33. 1979.

Nijhawan, P. 1981. Ventilation: Looking at Close Capture Ventilation Control. 33 Metal Producing. 19:FE54.

Ochs, J. T. 1975. Venting and Extraction in Electroplating Plants. Wasser, Luft and Betr. (Germany), 19(7):400-402.

Odasso, A. 1980. Air Pollution Abatement at the Brackenridge Works of Allegheny Ludlum Steel Corp. Iron and Steel Engineer, 57(6):51.

Patton, S. 1973. Hooded Coke Quenching System for Air Quality Control. Iron and Steel Engineer, 50(8):37.

Pfeiffer, W. 1982. Exhaust Air Quantities of Open Type Exhaust Hoods, Staub Reinhart Luft, 42(8):303-308.

Puri, R. 1974. System Design for Particulate Control in Woodworking Plants. Heating, Piping, and Air Conditioning, 46(9):54-58.

Randall, W. C., and E. W. Conover. 1931. Predetermining the Aeration of Industrial Buildings. ASHVE Transactions (American Society of Heating and Ventilating Engineers), Vol. 37.

Richter, G. 1975. Modern Ventilation Plant. The Current Technical Position in the Light of New Legislation on Protection of the Environment Ind.-Anz. (Germany), 97(75):1644-1648.

Rounds, W. P. 1977. Energy Considerations in Paper Machine Hood and Air Systems Design. Presented at TAPPI Engineering Conference, Atlanta, September 19-22, 1977. pp 149-153.

Rowe, A. D. 1970. Waste Gas Cleaning Systems for Large Capacity Basic Oxygen Furnaces. Iron and Steel Engineer, 47(1):74.

Rudolph, H. 1977. Engineering Criteria for a Hooded Quench Car System. Iron and Steel Engineer, 54(3):27.

Rudolph, H., and S. Sayer. 1977. Engineering Criteria for a Hooded Quench Car System. Iron and Steel Engineering, 54(3):27-32.

Sensenbaugh, J. D., and G. G. Persons. 1978. Electric Furnace Melt Shop Fume Control System at the Timken Co. Iron and Steel Engineering 55(10):53-55.

Shaughnessy, J. 1981. A Review of Shed and Gas Cleaning Systems for Controlling Coke Pushing Emissions from Coke Plants. In: Proceedings of the Symposium on Iron and Steel Pollution Abatement Technology for 1981, Chicago, Illinois, October 6-8, 1981. EPA-600/9-82-021.

Shcherbakov, K. L. 1976. Floor Layouts and Air Conditioning in Foundries for Mass and Large-Batch Production. Liteinde Proizyod (USSR), No. 10:34-5.

Siegel, M. E., and A. Shacter. 1978. Engineering Design Handbook for Air Cleaning For Chemical Demilitarization. ARCSL-CR-78022; AD-E410 041.

Spawn, D. 1981. Blast Furnace Cast House Control Technology and Recent Emissions Test Data for 1980-81. In: A Specialty Conference on Air Pollution Control in the Iron and Steel Industry, April 21-23, 1981.

Spawn, D. 1981. Field Evaluation of Fugitive Emissions from BOF Steelmaking Shops. In: Proceedings of the Symposium on Iron and Steel Abatement Technology for 1980, Philadelphia, PA. November 18-20, 1980. EPA-600/9-81-017.

Spawn, D. 1981. Status of Casthouse Control Technology in the United States, Canada, and West Germany in 1980. In: Proceedings of the Symposium on Iron and Steel Abatement Technology for 1980, Philadelphia, PA. November 18-20, 1980. EPA-600/9-81-017.

Squires, B. J. 1971. Electric Arc Furnace Fume Control and Gas Cleaning. Filter Separ., 7(4):447-450, 453-455.

Steiner, J., and L. F. Kertcher. 1980. Fugitive Particulate Emission Factors for BOP Operations. In: Proceedings of the First Symposium on Iron and Steel Pollution Abatement Technology, Chicago, Illinois, October 30-November 1, 1979. EPA-600/9-80-012.

Thaxton, L. A. 1971. Technology for Pollution Control. Exhaust Systems. Hood Close to Source Is Key Consideration. Metal Progr., 100(7):56-58.

Tsuji, K., T. Hayaski, and M. Shibatu. 1979. Required Exhaust Volume for Local Lateral Hoods. Bulletin, University of Osaka Prefect Ser. A., 27(1): 109-124.

Turner, H. 1981. Performance of BOF Emission Control Systems. In: Proceedings of the Symposium on Iron and Steel Pollution Abatement Technology for 1981, Chicago, Illinois. October 6-8, 1981.

Under EPA Duress, U.S. Steel Finds New Ways to Operate its Veteran Homestead OH Shop. 1979. 33 Metal Producing. pp 49-51.

U.S. EPA. 1974. Standards Support Document: An Investigation of the Best Systems of Emission Reduction for Electric Arc Furnaces in the Steel Industry. Draft.

U.S. EPA 1977. Blast Furnace Casthouse Control Technology Assessment. PB 276999.

U.S. EPA. 1980. Revised Standards for Basic Oxygen Process Furnaces - Background Information for Proposed Standards. Preliminary Draft.

U.S. EPA. Undated. An Investigation of the Best Systems of Emission Reduction for Pushing Operation of Byproduct Coke Ovens Standard Support and Environmental Impact Statement. Office of Air Quality Planning and Standards, Research Triangle Park, North Carolina.

Vatavuk, W. M., and R. B. Neveril. 1980. Estimating the Size and Cost of Pollutant Capture Hoods--3. Chemical Engineering, 87(24):111-112, 114-115.

Vaughan, J. 1975. A Residuals Management Model of the Iron and Steel Industry: Part I. Ph.D. Dissertation, Georgetown University.

Vinson, R. P., et al. 1980. SF6 Tracer Gas TEsts of Bagging-Machine Hood Enclosures. Bureau of Mines, Pittsburgh Research Center BUMINES-R1-8527.

Voldshina, R. K., and A. I. Sukhareva. 1972. Means of Reducing the Dust Concentration of the Air in Glass Plant Batching Departments. Glass and Coram. (USA), 29(9-10):585-586.

Wheeler, F. M., and A. G. W. L. Lamont. 1979. Current Trends in Electric Meltshop Design. Iron and Steelmaker. pp 36-43.

Whike, A. S. 1979. Energy Conservation and Pollution Control--Two Advantages of Coil Coating. In: Proceedings of the 14th International Society of Energy Conservation Engineers Conference, Boston, August 5-10, 1979.

Williams, A. E. 1982. New Developments in Coke-Oven Machinery Technology. Iron and Steel Engineer, 59(7):25+.

Zaryankin, A. E., and V. P. Zhilinskii. 1975. Analysis of Design of Soviet Exhaust Hoods of Steam Turbines and the Possibility of Lowering Their Resistance. Therm. Eng. (GB), 22(3):58-62.